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U. S. S. *RICHMOND.*

DESCRIPTION AND OFFICIAL TRIALS.

BY COMMANDER C. W. DENSMORE, UNITED STATES NAVY, MEMBER.

The U. S. S. *Richmond,* originally known as Scout Cruiser No. 9, is one of seven (7) scout cruisers authorized by Acts of Congress, 29 August 1916, and 4 March 1917, at a contract price, to include hull and machinery, of cost plus ten (10) per cent. A supplementary contract changed the contract price to cost plus fixed profit with a bonus saving clause. The contract for the *Richmond* was signed with the William Cramp & Sons Ship and Engine Building Company, Philadelphia, Penna., on 30 July 1917. This contract also called for the construction of Scout Cruiser No. 10, to which the name *Concord* was later assigned. The supplementary contract was signed 11 July 1919.

A later contract called for three (3) additional vessels of the same general characteristics. The keel of the *Richmond* was laid 16 February 1920, and the vessel launched on 29 September 1921, preliminary trials by the contractor were held off the Delaware Capes on May 1st to 3rd, 1923, and the official standardization machinery and maneuvering trials were held May 11 to 19th, 1923, off Rockland, Maine, and en route to the contractor's yard.

The delay in laying the keel was caused by the completion of the destroyer construction program then in full swing in the contractor's yard.

HULL DATA.

Length between perpendiculars, feet and inches	550–00
Length over all, feet and inches	555–06
Length on L. W. L., feet and inches	550–00
Beam, molded, feet and inches	54–10
Beam, over armor, feet and inches	55–04
Beam, on L. W. L., feet and inches	55–03¼
Draft, designed, forward, feet and inches	13–06⅝
Draft, designed, aft, feet and inches	13–05⅝
Displacement on designed draft, tons	7046
Displacement per inch, on L. W. L., tons	50.72
Coefficient of fineness, block	.605
Coefficient of fineness, midship section	.945
Coefficient of fineness, L. W. L.	.704
At designed draft, Area immersed midship section, square feet	702.4
Area, L. W. L., plane, square feet	21298.0
Area, wetted surface, square feet	33400.0
Engine room feed tanks, tons	27.32
Reserve feed tanks, tons	350.9
Fresh water (ship's) tanks, tons	29.0

GENERAL DESCRIPTION OF HULL.

The scout cruisers resemble, in general appearance, the later destroyers, the principal differences in outboard profile being the tubular tripod foremast carrying the lookout and fire control structures and the cutaway of upper deck at Frame No. 110, also the twin-gun mounts forward and aft.

THE UPPER DECK.

The upper deck extends from the bow to Frame No. 110, with the usual anchor gear forward, a twin 6-inch gun mount in light protective enclosure at Frame No. 18; 6-inch guns, starboard and port at Frame No. 28, admirals' and captains' cabins, staterooms, pantries and offices, boat stowage for all ship's boats, airplane catapults and airplane stowage, anti-aircraft guns, triple torpedo tubes, starboard and port at Frame No. 100.

From this deck rise structures for bridge, fire-room ventilators, radio room, and searchlight platforms abreast the after smoke pipes. The bridge structures consists of a platform deck from Frame No. 25 to Frame No. 39 carrying 6-inch guns starboard and port at Frame No. 28 or directly over those on the upper deck. Conning tower charthouse and two staterooms are also on this deck level. Above the conning tower is located the pilot house, navigating bridge and emergency cabin.

At the mast tripod junction is located the forward fire control tower, with searchlight and lookout platform just below. All boats stowed on this deck are hoisted on davits of the usual swivel type. Between Frames Nos. 103 to 110 a platform deck carries 6-inch guns starboard and port at Frame No. 106 1/2 and above this a range finder platform and after fire control station.

The mainmast carries booms for handling airplanes.

MAIN DECK.

On this deck is located lamp room, anchor windlass room, wardroom, officers' showers, staterooms, messroom, pantry, postoffice, ship's store, crew's space, general mess pantry and issuing room, galley and bakery, boiler room uptake spaces, fireroom and engine room access hatches, torpedo workshop, twin torpedo tubes starboard and port in enclosures at Frame No. 89, laundry, crew's washroom and lavatory, auxiliary radio

room, gasoline stowage tanks, 6-inch guns starboard and port at Frame No. 106 1/2, and twin 6-inch gun mount in protective enclosure at Frame No. 124.

FIRST PLATFORM DECK.

The first platform deck extends from bow to Frame No. 47 and from Frame No. 97 to stern. Forward is paint locker, chain locker, warrant officers' lavatory, ward room and warrant officers' staterooms, warrant officers' messroom, pantry, dispensary, dressing room, sick-bay, engineers' office, pay officer office, and clothing and small store issuing room. Aft is located crew space and chief petty officers' quarters.

SECOND PLATFORM DECK.

This deck extends from bow to Frame No. 34. Abaft the chain lockers are located supply department, storerooms, ordnance storeroom, sail locker, ship's stores storeroom and captain's storeroom. Deep fuel oil tanks extend from Frame No. 34 to Frame No. 46 1/2. Aft are located machine shop, evaporating and refrigerating machinery, wet and dry provision rooms, cold storage, main issuing room, after steering station and steering gear. Deep fuel oil tanks extend from Frame No. 97 to Frame No. 100 and from Frame No. 109 to Frame No. 117.

HOLD DECK.

On this deck are located peak tank, mine track stowage, ship's tanks, pumproom, magazines, wardroom and warrant officers' storerooms, central station, switchboard room, supply department stores. Fuel oil tanks extend from Frame No. 34 to Frame No. 46 1/2. The machinery spaces extend from Frame No. 47 to Frame No. 93, and the entire width of the vessel. Aft of Frame No. 93 are located fuel oil tanks, engineering and electrical storerooms, magazines and handling rooms, and after peak tank. Reserve feed water tanks are located under the boiler rooms, *i.e.* from Frame No. 47 to Frame No. 59 and from Frame No. 70 to Frame No. 82.

COMPLEMENT

Admiral	1	
Staff	6	
		= 7
Commanding Officer	1	
Wardroom Officers	21	
Warrant Officers	10	
		=32
Total Officers		39
Seamen branch	209	
Artificer branch	33	
Engine room force	132	
Commissary branch	10	
Special branch, yeoman, storekeepers, pharmacists, hospital apprentices, bugler, etc.	18	
Messman branch	17	
Total crew	419	

BATTERY.

The main battery consists of twelve (12) 6-inch—53 caliber guns mounted in locations previously noted. These guns are served by six (6) ammunition hoists of the step by step type, operated by constant speed electric motors of 7 1/2 H.P., working through Waterbury variable speed gears. The maximum rate of delivery is ten (10) rounds per minute. The hoists for the broadside guns, deliver on both the main and upper decks. The hoists for the twin mount guns deliver at the base of the mount, where transfer is made to endless chain hoists that serve the guns individually. The anti-aircraft battery consists of four (4) 3-inch—50 caliber guns, served by band operated hoists. Two (2) 3-pound guns for saluting purposes are supplied. There are two (2) triple torpedo tubes on upper deck

and two (2) twin tubes on main deck. Four (4) reload torpedoes are carried. There are two (2) airplane launching catapults on the upper deck and two (2) scouting planes are carried.

SMALL BOATS.

1—36-foot motor sailing launch.
2—35-foot motor boats.
1—30-foot motor sailing launch.
2—30-foot whaleboats.
2—16-foot dinghys.
2—12-foot punts.

All boats hoist on davits of the usual swivel type, the dinghys stowing in the 30-foot motor sailing launch.

ANCHORS, ANCHOR ENGINE AND GEAR.

The windlass was constructed by the American Engineering Company, Philadelphia, Penna., and is of the vertical type operated by a two (2) cylinder steam engine, having 14-inch diameter cylinders by 12-inch stroke. An extension shaft drives a worm engaging two worm wheels at the lower ends of the wildcat shafts. The locking gear and friction band brakes are located in the anchor engine room. In operation, the worm and worm shaft thrust are flooded with oil supplied by a small gear pump directly connected to the forward end of the engine crank shaft. Deck control of the engine is provided for, reversing being accomplished by a master valve, also deck operated. An extension of the worm shaft, through suitable gearing, drives two capstan heads on deck, for anchor handling and docking purposes. The wildcats are arranged to take a 2 1/2-inch stud link cable, 120 and 180 fathoms being provided on the starboard and port sides respectively. The two bow anchors are of the stockless type, 11,000 pounds each, arranged to stow clear of the side. A 3,000 pound stockless stem anchor is provided, also a Navy type Kedge of 400 pounds.

DECK WINCHES.

There are two (2) steam operated, double gypsy, compound geared winches built by the American Engineering Company, Philadelphia, Pa., on the upper deck, for boat hoisting and general service, also a small electrical driven mine handling winch on main deck at the stem. The upper deck winches have the following capacities:

Forward Winch—Double gypsy heads, 10,000 pounds at 100 feet per minute on high gear, or 20,000 pounds at 50 feet per minute on low gear.

After Winch—Double cable drums, double gypsy heads, 6,000 pounds at 180 feet per minute on high gear, or 21,600 pounds at 50 feet per minute on low gear.

STEERING GEAR.

The steering gear consists of an electro-hydraulic system built by the American Engineering Company, Philadelphia, Pa., and having two pairs of opposed hydraulic cylinders directly connected to the rudder yoke by connecting rods. Oil is pumped from one hydraulic cylinder to the other by variable stroke pumps of the Waterbury Tool Company's type. These pumps, in duplicate, are operated by 60 H.P., shunt wound, constant speed motors. A distributing valve controls the oil flow from either pump to the cylinders, and cutout valves allow the use of all or any selected pair of hydraulic cylinders. Relief valves set to operate at 2,000 pounds per square inch prevent excess pressures in the various lines. The amount of oil pumped and the direction of its flow is dependent upon the position of the tilting plate in the Waterbury gear. This plate is, in turn, moved by one of the two 3 H.P. compound wound pilot motors, controlled from the steering stations forward. The movement of the tilting plate by the pilot motor is through a floating lever, so connected that the consequent motion of the rudder tends to return the tilting plate to the zero or neutral position. A torque equalizing, or pressure control system is

incorporated in the follow-up mechanism to prevent stalling motors by overloading and to prevent the rudder from taking charge. The pilot motor is started, stopped or reversed by a standard type controllers of the non-follow-up type located at the various steering stations. An electrically operated band brake working on a drum on the pilot motor shaft prevents over-running. Suitable hand operated clutches are provided, which, in conjunction with switches allow either of the pilot motors to be operated from any of the various control stations. A "trick" wheel is also provided to allow manual operation of the tilting plates for test purposes or in case of emergency. An interlocking arrangement prevents putting both pilot motors in gear at the same time.

Auxiliary steering arrangements consists of a Waterbury Tool Company's variable stroke pump operated by the usual type of triple hand wheel. The tilting plate of this auxiliary pump is manually operated.

LAUNDRY.

The laundry is located on the main deck and is equipped with the following apparatus:

1—Washing machine having two 24 × 36-inch diameter compartments, made by The Henrici Laundry Machinery Company, Boston, Mass., driven by a 2 H.P. compound wound General Electric Company's motor.

1—26-inch Centrifugal Extractor, made by American Laundry Machinery Company, New York, N. Y., driven by a 2 H.P. Diehl compound wound motor.

1—Tumble Drier, American Laundry Machinery Company's Vento type, driven by a 3 H.P. Diehl shunt wound motor.

1—48-inch steam heated flat work ironer, made by the American Laundry Machinery Company and driven by a 1 H.P. General Electric Company's shunt wound motor.

1—Body Ironer, steam heated, made by the Daly Manufacturing Company, Cincinnati, Ohio.

1—Collar Presser, made by American Laundry Machinery Company.

1—Navy type steam starching cauldron, 17 × 14 inches.

1—Tub.

GALLEY AND BAKERY.

In the galley there are five (5) 3 1/2-foot sections, Type A, Navy Standard oil burning ranges, built by the S. B. Sexton Stove and Manufacturing Company, of New York. Atomization is by air. A fifty gallon oil tank feeding the ranges is located in starboard passage way abreast the galley and is connected to the booster pump discharge for filling, the oil being forced to the burners from the tank by air pressure. A motor driven single stage, double cylinder air compressor with a capacity of fifty cubic feet of free air per minute, built by the Allis-Chalmers Manufacturing Company, Milwaukee, Wis., is installed in the forward engine room, for supplying air to the ranges at 20 pounds pressure per square inch. An automatic pressure control system is provided.

The following additional equipment is installed:

4—60 gallon and 1—40 gallon aluminum steam jacketed kettles, made by the Aluminum Cooking Utensil Co., New Kensington, Penna.

1—Blakeslee potato and vegetable peeler, driven by a Robbins and Myers 1/2 H.P. shunt wound motor.

1—Meat chopper, made by The Enterprise Manufacturing Company, Philadelphia, Penna., driven by a 3/4 H.P. Diehl compound wound motor.

1—Dough mixer, made by the J. H. Day Company, Cincinnati, Ohio, driven by a 1 H.P. General Electric Company's shunt wound motor.

1—No. 2 size Navy Standard Baker Proofer, steam heated.

1—Four deck, No. 2 type, Navy Standard electric bake oven, made by the Edison Electric Appliance Company of New York, with a maximum current consumption of 11,000 watts.

1—Warming oven, steam heated. Overhead racks for galley utensils are provided, also a table and sink, with built-in drawers and lockers underneath.

SCULLERY.

The scullery is located on starboard side of main deck just aft of No. 2 fireroom uptake space and is fitted with a Navy Standard dish washing machine, supplied by The Insinger Company, Philadelphia, Pa. This machine is electrically operated by a 3/4 H.P. compound wound motor built by The Star Electric Motor Company, of Newark, N. J. Sink and shelves are provided for storage of all crews' dishes and cutlery.

DRAINAGE SYSTEM.

The drainage system consists of a five (5) inch line from the forward fire room to the after engine room, from which points smaller leads run to the forward and after peak tanks. This line is connected to one fire and bilge pump in No. 2 and one in No. 3 firerooms; also to two fire and bilge pumps in each of the forward and after engine rooms. Manifolds at Frames 14, 24 and 34 take care of compartments forward, while manifolds at Frames Nos. 98 1/2 and take care of compartments aft.

Arrangements are made for sluicing water from steering gear and hand steering gear compartments to after peak tank, from which it may be removed through a suction pipe on second platform deck by a portable electrically driven pump.

FLOODING SYSTEM.

Magazine flooding is arranged for the forward and after groups of magazines and shell rooms, with ten (10) and eight (8) inch sea valves respectively.

Flooding the peak tanks is by means of fire hose connected to the nearest plug.

A magazine sprinkling system is provided, supplied from the fire main.

FIRE PROTECTION SYSTEM.

This may properly be divided into two separate systems, depending on the character of the fire *i.e.*:

1—Water.

2—Foam, for oil fires.

(1)—This consists of a 4 1/2-inch lead lined fire main running the entire length of the vessel, with suitable outlets in all compartments, and on main and upper decks. This fire main has connections to the fire and bilge pumps in engine rooms and fire rooms. Connections are provided to the magazine sprinkling systems, stern tube flushing line, dynamo oil coolers, and through a reducing valve to the sanitary line. Cut-out and relief valves are provided at suitable points.

(2)—A foam fire extinguisher system is provided for use against oil fires in the fire rooms.

VENTILATION AND HEATING SYSTEM.

To provide ventilation throughout the ship twenty-one (21) supply and exhaust ventilating sets, nine (9) of which have heating coils, are installed, each set consisting of an electric driven centrifugal multivane type fan, piped to a system of ducts from which branches lead to the various compartments requiring artificial ventilation. Where heating is required, the air from the fans is passed through steam heated coils before delivery to the air ducts. Regulation is by means of steam in the coils, also by means of by-pass control dampers in the air ducts. Means are also provided by which the humidity may be regulated through the operation of a hygrostat admitting live steam into the ventilating air. Three (3) 1/4 H.P. ventilating sets provide ventilation for the sound proof radio booths.

Certain places not requiring artificial ventilation are heated by the usual type of steam coil. Fire control and lookout stations are provided with electric heating units.

SANITARY SYSTEM.

The sanitary system consists of branches taken from the fire main, reducing valves being provided to assure proper pressures.

FUEL OIL FILLING AND STORAGE.

Fueling is accomplished through six (6) inch Tee valve connections at Frames Nos. 44, 53 and 93 on upper deck and Frame No. 110, main deck.

The connections at Frames Nos. 44 and 110 lead direct to manifolds for filling the forward and after deep tanks, while the connections at Frames Nos. 53 and 93 lead to manifolds whereby any one of the tanks may be filled.

Each oil tank is provided with suitable steam coils.

A pneumercator system is provided, the gauges for the forward group of tanks being in the air lock Frame No. 53, starboard side of No. 1 fire room, while the after group gauges are located on port side of after engine room, Frame No. 93.

Each fuel oil tank is provided with an air vent pipe fitted with De Craft non-return vent valves. Sounding pipes are also provided for each tank.

GASOLINE STOWAGE.

Stowage for gasoline is provided in two vertical cylindrical tanks located on Main Deck at Frames Nos. 106–109 1/2. A pressure of 10–15 pounds per square inch is maintained in these tanks with $C\ O^2$ gas, and means are provided for quickly emptying them overboard in case of fire hazard.

COMPRESSED AIR SYSTEM.

A compressed air service, consisting of an air line with branches to practically all compartments of the vessel, furnishes air at 150 pounds pressure per square inch for the following purposes:

Gas ejecting at guns, pneumatic tube system, pneumatic tools in machinery spaces, burner and boiler cleaning in firerooms, pneumercator gauge boards in firerooms and engine rooms, Hygrostats for humidity control on heating and ventilating systems, airplane catapults, galley oil burning system.

This line is supplied by two single stage steam driven, 11 × 11 × 12 inch compressors, made by the Westinghouse Air Brake Company, Pittsburgh, Penna., and located in No. 2 and No. 4 firerooms respectively. Each compressor has a storage tank of sixteen (16) cubic feet capacity. (See Plate II—11).

There are two torpedo air compressors, located in the port forward corner of the forward engine room. They are of the four stage vertical steam driven type built by the Platt Iron Works Company, Dayton, Ohio. (See Plate III—28).

Each compressor will deliver 20 cubic feet of air at 3,000 pounds pressure per square inch per hour when running at 350 revolutions per minute.

Two groups of sixteen flasks each are provided for air storage and a reducing valve is fitted for supplying the ships pneumatic service main.

FRESH WATER SYSTEM.

There are two ships tanks, A 104—105, having a capacity of 7,500 gallons or 29 tons. Water is distributed by two, two stage centrifugal pumps, made by the Morris Machine Works and driven at 1,600 to 1,850 R.P.M., by Westinghouse compound wound motors, through a 2 1/2 inch fresh water line, from which branches are to supply officers staterooms, bathrooms, galley, scullery, laundry, crews washroom, and scuttlebutts. Neptune Meter Company's "Trident" meters are provided for checking water consumption at various points.

Reserve boiler feed is carried in the following compartments:

Compartment	*Gallons*
B 1	9,825
B 2	9,825
B 3	10,430
B 4	10,030
B 5	13,544
B 6	13,544
B 7	13,260
B 8	13,260
Total	93,718 Gal. = 350.9 Tons

Suction from these tanks may be taken by the auxiliary feed pumps or by the make up feed pumps.

The engine room feed tanks have a capacity of 3,674 gallons or 13.66 tons each.

REFRIGERATING SYSTEM.

The refrigerating system is located on the port side of the second platform deck between Frame Nos. 93-97, and consists of two type 6 E, two ton ethyl-chloride machines made by the Clothel Company of New York. They are self contained units and are driven by a 10 H.P. shunt motor made by the Electric Dynamic Company of Bayonne, New Jersey. There are three cold storage rooms, for meats, vegetables and officers stores, and an ice making box located on main deck at Frame No. 95, having a capacity of six (6) cakes of standard size. Cold brine is also supplied to the cooling coil in the crews scuttlebutt, starboard side at Frame No. 96.

EVAPORATING PLANT.

The evaporating plant is located on the starboard side of the second platform deck between Frames Nos. 93-97, and consists of four (4), size 28, units of the Reilly Submerged coil type of evaporators, made by the Griscom-Russell Company of

New York, arranged for compound effect with a total rated capacity of 25,000 gallons of potable water per twenty-four hours when operated with 70 pounds of steam in the 1st effect coils, with a 40 per cent capacity increase with clean coils.

A connection is also provided for using exhaust steam in the first effect coils, which will result in considerable economy, although a somewhat reduced distillate capacity.

There are two (2) distillers of the curved tube condenser type, each distiller having 346 5/8-inch diameter tubes 4 inches long between tube sheets giving 226.45 square feet cooling surface. There is one (1), 6 1/2 × 10 × 8 inches simplex air pump of the Blake-Knowles type built by the Worthington Pump and Machinery Corp., for handling the distillate and maintaining a fairly high vacuum on the distillers.

Cooling water for the distillers is supplied by two (2) five (5) inch volute Worthington centrifugal pumps driven by Sturtevant steam turbines.

Sea water is supplied to the evaporators by two (2) 4 1/2 × 6 × 6 inches Simplex pumps made by the Worthington Company, who also supplied the two (2) 4 1/2 × 5 × 6 inches Simplex fresh water pumps, and the 4 1/2 × 6 × 6 inches Simplex brine pump. There are two 100 gallon testing tanks, which may be alternately filled, tested, and distributed to ship's tanks or reserve feed tanks as needed.

Separators are provided in the vapor line from each evaporator shell.

There are two (2) vapor feed heaters of 23.93 square feet heating surface fitted with 5/8 inch diameter U shaped tubes, and one (1) coil drain heater also of the U-tube type and having a heating surface of 18.3 square feet.

MACHINE SHOP EQUIPMENT.

The machine shop is located on the second platform deck between the spaces occupied by the refrigerating and evaporating plants, and consists of the following machines, all with independent electric drive as indicated:

Machine	Motor
Gap Lathe—16 × 48 inches, 32-inch swing in gap, made by E. Harrington Son and Company, Philadelphia, Pa.	Westinghouse 2½-H. P. shunt.
Tool Room Lathe—12 × 48 inches, with taper attachment, built by Reed and Prentice Company, Worcester, Mass.	Westinghouse 2-H. P. shunt.
Milling Machine—No. 1 Universal, made by the Garvin Machine Works, New York, N. Y.	Westinghouse 2-H. P. shunt.
Drilling Machine—14-inch pillar type, W. E. & J. Barnes Company, Rockford, Ill.	Westinghouse 2-H. P. shunt.
Shaper—16-inch backed geared, made by Cincinnati Shaper Company.	Westinghouse 2-H. P. shunt.
Grinder—Type D. F. G., direct drive motor with two (2) 12 × 2-inch wheels, made by the Willey Electric Company, Louisville, Ky.	

A twelve foot workbench is provided with suitable drawers, lockers, etc., for tools and supplies.

GENERATING PLANT.

The generating plant consists of four (4) Westinghouse 100 K.W. horizontal indirect connected, 125 volt generators, driven by a single wheel impulse type turbines, there being two (2) generators in each engine room. (See Plate III—21.)

The turbine wheel has a single row of blades. The steam expands in the nozzles where the heat energy is changed to velocity energy, then goes through the blades where part of the velocity energy is changed into work, then through a reversing chamber and after a second pass through the wheel the steam goes to the exhaust.

The rotor is an unperforated disc with the blades secured by pins in an annular groove. The blade shrouding is integral with the blades.

The rotor is held between two lengths of flanged shaft by bolts. The rotor has 1/32 inch axial clearance from steam nozzles and 1/16 inch clearance from reverse chambers, while the radial clearance is 3/16 inch.

Provision is made for preventing any danger of shaft breaking or damage to the turbine cylinder by over-speeding.

Water sealed glands are provided to prevent air or steam leakage where the turbine shaft passes through the cylinder, and a steam seal gland is fitted for use when starting and until sufficient speed is attained to make the water seal operate or in case the water supply fails when running condensing. An overspeed automatic stop governor is fitted on the end of the turbine shaft, the spring retained plunger of which is set to operate at about 10 per cent overspeed.

An over-pressure stop automatically closes the throttle if the pressure in the turbine casing exceeds a predetermined amount. The turbine casing is also provided with a 1/2-inch signal valve set to operate at 15-20 pounds back pressure, also a relief valve of sufficient size to protect the casing, under the extreme condition of 200 pound steam and exhaust valve closed. A circuit breaker trip switch operated automatically when the throttle valve is within 1/8 inch of its seat, actuates the switchboard circuit breaker tripping magnet, to prevent motorizing in case of a reverse current.

The reduction gear is in a separate case between the turbine and generator and consists of the pinion, gear wheel, bearings, governor and oil pump. Both the turbine and gear cases have horizontal joints. The gears are of the herring bone type with involute teeth. The turbine and pinion shaft are connected by a solid muff coupling, while the gear and generator shafts are connected by bolts in a solid flange coupling.

The vertical fly ball governor is driven from the inner end of the gear shaft, by skew gears. On the lower end of the governor shaft is mounted the gear type oil pump which runs submerged in oil. The governor and oil pump are carried in a single bracket and may be lifted out together for inspection and

repairs. The lower part of the gear case serves as the oil reservoir, from which the oil is forced through the external cooler and duplex strainer, to the open strainer on top of the gear case, thence to the gear, pinion, turbine and generator bearings and back to the reservoir.

The set is self lubricating at all speeds above one fourth speed. A hand oil pump is provided for use when starting up. The external oil cooler is made by the Globe Engineering Company, Philadelphia, Penna., and is of the straight tube type, having nineteen 5/8 inch O.D. tubes, 36 inches between tube sheets giving 9.35 square feet cooling surface.

GENERATOR.

The generator frame is of cast steel and split horizontally. There are six laminated steel main poles and six solid steel commutating pole pieces, so arranged that any pole piece and its field coil may be removed without disturbing the armature.

The machine has shunt, series and commutating pole windings.

The armature core is of the laminated drum type with open slots. The laminations are pressed and keyed onto the shaft and riveted between pressed on end plates.

The commutator is composed of 171 hard drawn copper bars, insulated from each other by .013 inch mica plates. The commutator is 12 1/2 inches diameter, 13 1/8 inches long and has a wearing depth of 3/4 inch.

The armature coils are form-wound of D.C.C. copper strap and are held in place in the slots by fibre wedges and coil ends are held down by wire banding.

There are six brush holder rods each carrying six (6) 5/8-inch × 1-inch brushes which are held on the commutator by steel springs adjustable as to tension.

Four terminals are mounted on an ebony asbestos terminal board insulated from the generator frame, and provide for two standard Navy 650,000 C.M. cable for each main lead and one equalized cable of equal cross-section.

A field rheostat of the face plate type is provided having a total resistance of 22.7 Ohms which is sufficient to reduce the 'full load' voltage to 40 per cent below normal and the 'no load' voltage to approximately zero.

The generator shunt current at hot full load and no load is 5.7 ampères.

SWITCHBOARDS.

There is one main switchboard in each engine room, interconnected and supplying power and lighting distribution panels located in various parts of the vessel. The main switchboards are fitted with the usual voltmeters, ammeters, main and equalizer switches, power and lighting distribution switches, rheostats, etc., and distribute power and lighting to the following motors and power panels:

Forward engine room, first platform, Frame No. 69-P.	For power	Power panels Nos. 1, 3, 7, 11, 12, and 13; forward controller steering gear; interior communication switchboard and battery-charging panel; 20 K. W. radio motor generator; ventilation sets Nos. 16 and 17; main radio room; forward turret motors; power to after switchboard; shore connection.
do	For lighting	Lighting distribution panels Nos. 1 and 2; Nos. 1 and 2 36-inch searchlights; also 24-inch navigating searchlight; port and starboard 12-inch signal searchlights. Running and battle lights forward; lighting, forward and after generators; lighting bus to after switchboard.
After engine room, first platform, Frame No. 92-S.	For power	Power panels Nos. 2, 4, 5, 6, 7, 8, 14, and 15. After controller steering gear; 2 K. W. radio motor generator; ventilating sets Nos. 18 and 19; mine handling winch; main radio room; after turret motors; power to forward switchboard.

After engine room, first platform, Frame No. 92-S.	For lighting	Lighting distribution panels Nos. 3 and 4; Nos. 3 and 4 36-inch searchlights; battle lighting aft; general night lighting aft; lighting forward and after generators; lighting bus to forward switchboard.

The following list of power panels shows location and motors served:

Panel	Location	Supply
1	Forward engine room, Frame No. 70-P	Port and starboard turbo turning motors; turbine gland leak-off motor; oil separator motor.
2	After engine room, Frame No. 93-S	Port and starboard turbo turning motors; turbine gland leak-off motor; oil separator motor.
3	Second platform, Frame No. 32-P	Nos. 1, 2, and 3 ammunition hoist motors.
4	Second platform, Frame No. 95-A	All workshop tools.
5	Second platform, Frame No. 93-P	Refrigerating plant motors.
6	Second platform, Frame No. 106-S	Nos. 4, 5, and 6 ammunition hoist motors.
7	Second platform, Frame No. 127-A	Panels Nos. 9 and 10.
8	Second platform, Frame No. 127-A	Steering gear pilot motors.
9	Second platform, Frame No. 127-S	Starboard steering gear motor.
10	Second platform, Frame No. 127-P	Port steering gear motor.
11	Main deck, Frame No. 23-P	Ventilating sets Nos. 1, 3, and 11; fresh water pumps.
12	Main deck, Frame No. 43-P	Ventilating sets Nos. 4, 12, and 13; sounding machine motors; forward fire control station heaters.

Panel	Location	Supply
13	Main deck, Frame No. 70-P	Ventilating sets Nos. 2 and 5; galley bake oven; meat grinder; dough mixer; potato peeler; dish washer; air compressor.
14	Main deck, Frame No. 95-S	All laundry machines.
15	Main deck, Frame No. 100-P	Ventilating sets Nos. 6, 7, 8, 9, 10, 14, 15, 20, and 21; auxiliary radio booth; ventilation motor.
Interior Communication Panel	Hold, Frame No. 27-P	Fire control telephone talking; interior communication motor generators; gun firing motor generators; telephone motor generators; gyro compass; follow-the-pointer motor generator; stabilized sight motor generator.
Lighting Distribution Panel No. 1	Main deck, Frame No. 41-A	Bridge, upper, main, first, and second platform decks from bow to Frame No. 61.
2	Hold, Frame No. 31-P	All navigation; anchor; signal and fighting lights; fire control and foremast searchlight platform.
3	Main deck, Frame No. 93-A	Upper and main decks. Frame Nos. 64–110.
4	First platform, Frame No. 101-A	Main, first, and second platform. Frame Nos. 101 to 125.

INTERIOR COMMUNICATION SYSTEM.

The interior communication system comprises an interior communication switchboard located in Interior Communication Room hold deck, Frames Nos. 26–34–P, which is energized from the forward main switchboard. The 20 Volt bus bars of this switchboard are fed either by motor generators or storage batteries and supply all the call bell, telephone and transmitter-indicator systems.

Circuit	Supply
A	Admiral, captain, wardroom and warrant officers' quarters.
BD	Follow-the-pointer system.
BR	Lifebuoy release system.
E	All voice tubes except fire control.
EB	Fire room feed signal system.
F	Fire alarm system.
FT	Time of flight warning signal.
G	General alarm gong system.
H	Boat hour gongs.
HC	Anemometer indicator circuit.
J	Ship's service telephones.
JA, 2JW	Fire control telephone system.
K	Shaft revolution indicator system.
KA	Secondary counter system.
L	Steering telegraph system.
LB	Steering emergency system.
LC	Gyro compass system.
1M, 2M, 3M, 4M	Engine revolution telegraph.
1MB, 2MB, 3MB, 4MB	Engine order telegraph.
MT	Mine laying timing system.
MV	Sound receiving system.
1N	Rudder indicator system.
2N	Auxiliary rudder indicator system
PA	Gun firing system.
QA	Air lock indicator system.
RD	Battle order indicator system.
SA	Smoke telegraph system.
SO	Submarine oscillator system.
1T, 2T, 3T, 4T	Torpedo firing and indicator system.
4TF	Torpedo target bearing indicator system.
TL	Dead reckoning tracer system.
1VB, 2VB	6-inch gun and anti-aircraft gun salvo signal system.
VD	Turret salvo buzzer system.
W	Electric whistle control.
X	Pneumercator system.
Y	Forbes log system.

MACHINERY LAYOUT.

GENERAL.

The machinery layout of the U. S. S. *Richmond* follows past Destroyer practice in general, except that the boiler and engine groups are alternated and we have a boiler group of six (6) boilers in two separate firerooms, an engine group driving the outboard shafts, next another boiler group and another engine group driving the inboard shafts, as shown on Plate I.

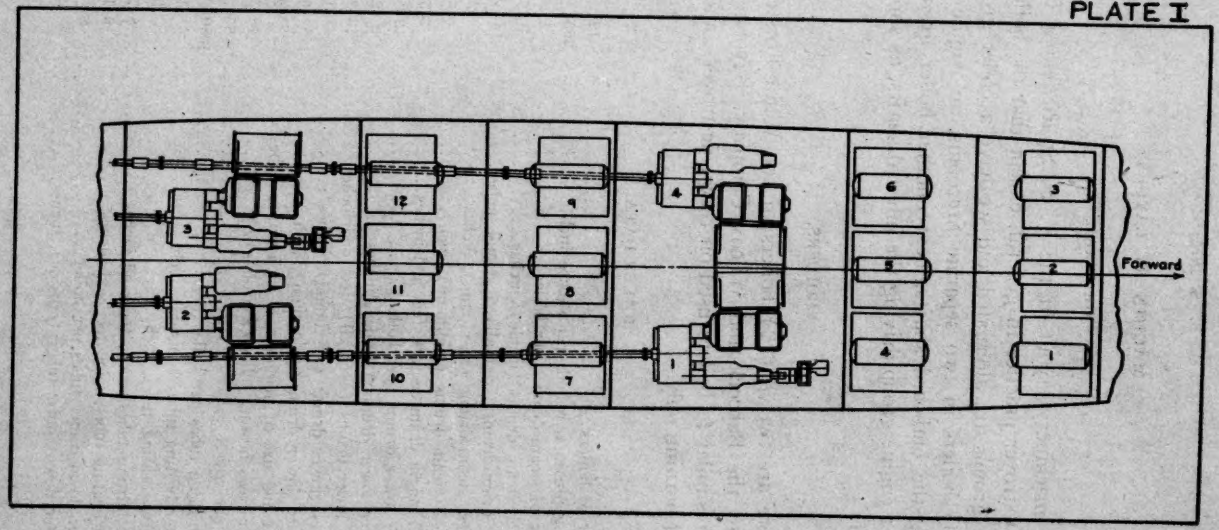
BOILERS.

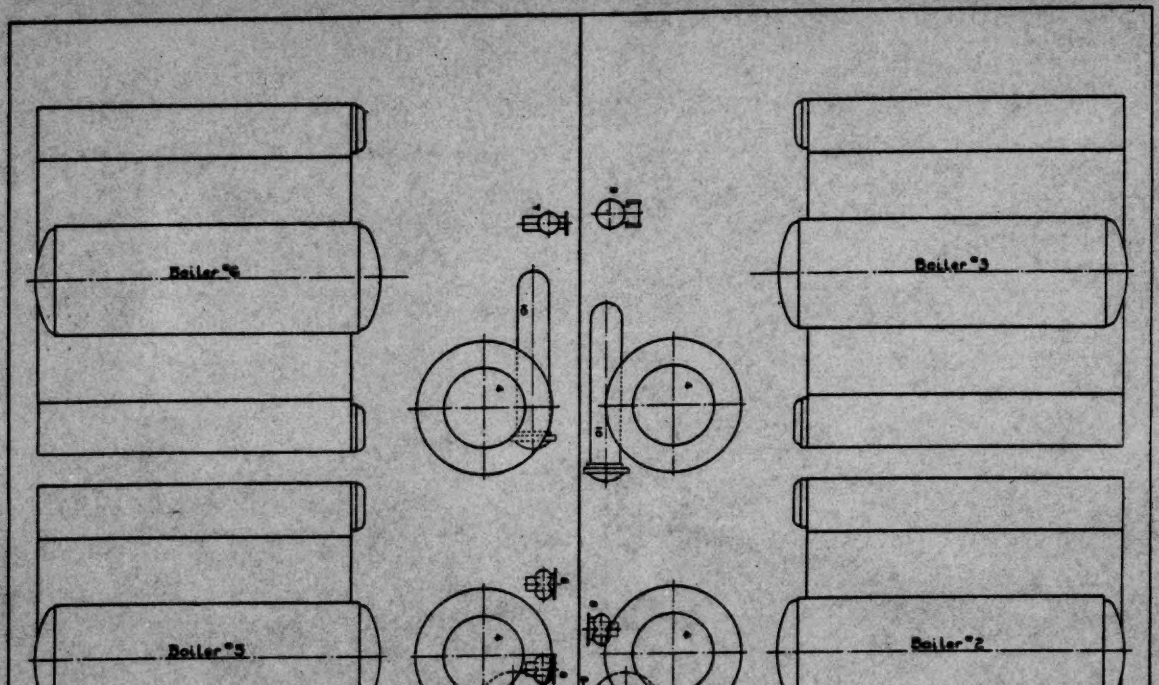
There are twelve (12) boilers of the White-Forster type built by the Babcock and Wilcox Company, Bayonne, New Jersey, similar to boilers used on recent destroyers, and fitted for oil burning only.

BOILER DATA.

Number of boilers	12
Working pressure, pounds per square inch	265
Number of steam drums (per boiler)	1
Diameter steam drums (inside), inches	54
Length steam drums (over all), feet and inches	14–5⅜
Thickness steam drums (maximum), inches	2
Thickness steam drums (minimum), inch	¾
Vertical height drum centers, feet and inches	7–5
Number lower drums (per boiler)	2
Diameter lower drums (inside), inches	24
Length lower drums (over all), feet and inches	13–2½
Thickness lower drums (maximum), inches	1¼
Thickness lower drums (minimum), inch	9/16
Number and size of manholes in each drum, inches	2, 12 × 16
Number rows heating tubes (2 groups)	48
Tubes per row	85
Total heating tubes per boiler	4080
Diameter heating tubes (outside), inch	1
Thickness heating tubes, inch	.095
Length heating tubes, max., feet and inches	8–7¾
Length heating tubes, min., feet and inches	6–10¾
Number down comer tubes per boiler	8
Diameter down comer tubes (outside), inches	4

PLATE I

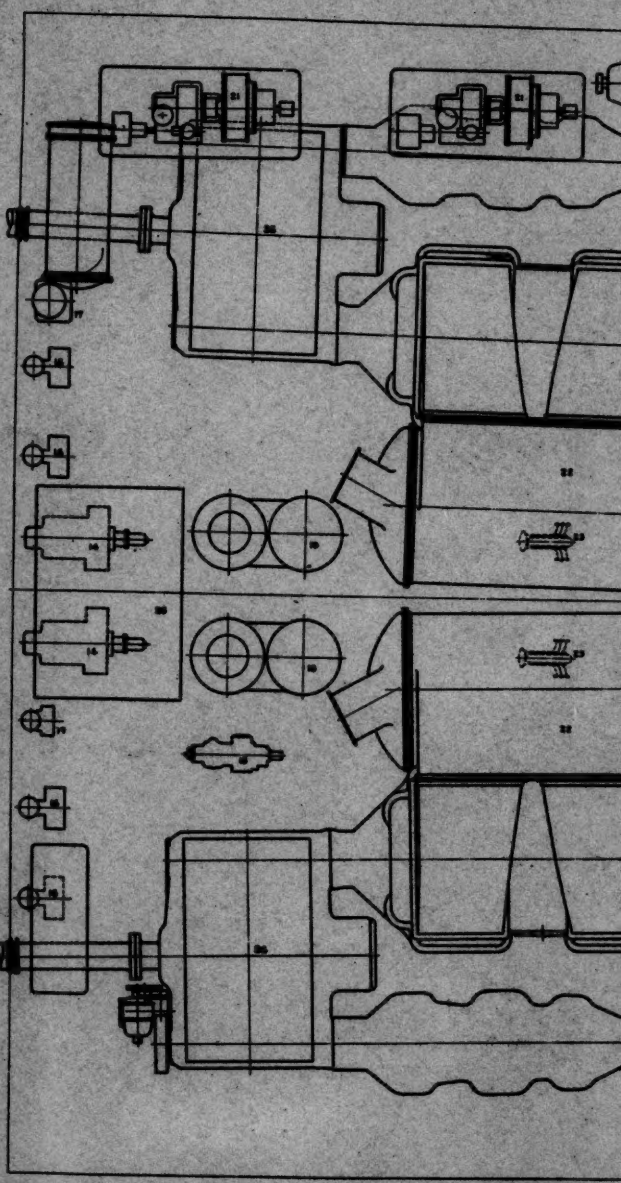




Layout of Forma

PLATE II

Layout of Forward Boiler Rooms



Layout of Forward Machinery Compartment

PLATE III

Compartment

Thickness down comer tubes, inch	.165
Heating surface per boiler, square feet	7570
Furnace volume, Nos. 2 and 4 to 12, cubic feet	695
Furnace volume, Nos. 1 and 3, cubic feet	677
Diameter main stop valve, inches	8
Diameter feed connections, inches	3
Diameter bottom blow connections, inches	1½
Diameter surface blow connection, inches	1¼
Number safety valve outlets	2
Diameter safety valve outlets, inches	6
Safety valves per boiler	2
Safety valves, type and size, twin, inches	4
Number burners per boiler	14
Diameter register opening, inches	10½
Type register, Bu. Engineering	Cone
Type atomizer, Bu. Engineering	Cone

The boiler furnaces are lined with 9 × 4 1/2 × 2 1/2 inch fire brick, backed with 1/4 inch sheet asbestos and 1 inch nonpareil H.P. blocks and cased with No. 7, B.W.G. (188 mils) plate. The casings above the tubes are of No. 14 B.W.G. (78 mils) plate lined with 1 1/4 inch nonpareil H.P. blocks held in place by a No. 14 B.W.G. (78 mils) plate, which is protected from direct contact with flame by a 1/4 inch asbestos sheet.

The furnace bottom is lined with 2 inch calcined sil-o-cel Grade C-3, and two layers of split fire brick 9 × 4 1/2 × 1 1/4 inch breaking joints and laid in fire clay and cement.

FORCED DRAFT SYSTEM.

There are twelve (12) forced draft blowers, three (3) in each fireroom located just below the main deck, (See Plate II-4) taking suction through separate trunks opening into the space between the boiler uptake spaces. Screened and hooded upper deck erections at Frames Nos. 53 and 76 provide fresh air intakes, while doors leading to the boiler uptake spaces and to the main deck living compartments provide for suction from these compartments. Air trunks in the forward end of each engine room opening into the boiler uptake spaces, provide

means for taking hot air from the engine rooms and assist in the ventilation of these spaces.

The forced draft blowers are of the vertical steam turbine driven type in common use in the larger Destroyers.

BLOWER DATA.

Maker, B. F. Sturtevant Company, Boston, Mass.	
Size and type	VD Type 7
Diameter bucket wheel, feet and inches	2–6
Width buckets, inches	2¾
Number buckets	80
Number nozzles and reverse buckets	8
Diameter nozzle throat, inch	11/32
Area nozzle throat, square inch	.09281
Nozzle mouth, inch, square	½
Number of reversals in each reverse bucket	4
Clearance between bucket and nozzles, inch	1/32
Diameter steam inlet to governor, inches	2
Diameter exhaust outlet, inches	4
Diameter relief valve, inches	2½
Type of governor, spring loaded fly ball, speed regulation.	
Bearing lubrication, self-contained forced feed.	
Lubricating oil pump	Gear
Type of fan	Turbo-vane
Diameter fan wheel, inches	48
Designed—	
Steam pressure, pounds per square inch	175 to 300
Exhaust, pounds obsolete	0 to 30
Revolutions per minute, normal	1350
Revolutions per minute, maximum	1700
Horsepower (normal R. P. M.)	100
Steam consumption, pounds per B. H. P. hour	62½
Capacity fan, cubic feet per minute	50000
Static head, inches, water	7

FEED WATER HEATER.

There is one (1) exhaust steam feed water heater in each fireroom, with piping so arranged that feed water may be sent through each pair in series, or only a single heater used. They have steel plate shells, steel tube sheets, composition exhaust nozzles and cast steel water bonnets. Baffles in the bonnets

compel four (4) passes through the tubes before the feed water reaches the outlet. Tubes are curved and expanded into tube sheets at both ends. (See Plate II-9.)

Diameter tube sheets, feet and inches	2–11½
Thickness tube sheets, inches	1⅜
Length between tube sheets, feet and inches	6–1
Diameter exhaust connection, inches	9
Diameter water inlet and outlet, inches	5
Diameter drain line connection, inches	3¾
Number tubes	680
Diameter tubes (outside), inch	⅝
Pitch of tubes, inch	29/32
Thickness of tubes, mils	49
Material in tubes	Comp. "A"
Total heating surface (1 heater), square feet	678.61

FUEL HANDLING SYSTEM.

The fuel handling system follows recent Destroyer practice; *i. e.* the fuel oil is taken from the storage tanks by the fuel oil booster pumps, (See Pump List for type and size) which discharge to the suction line of the fuel oil service pumps, which force the oil through oil meters, strainers and heaters to the burner lines on boiler fronts.

The fuel oil service pumps can also take suction direct from the fuel oil storage tanks.

The strainers are 2½ inch duplex type, with strainer baskets easily removable for cleaning.

There are two (2) U tube type fuel oil heaters in each fireroom. (See Plate II-10.)

Shell	Wrought steel
Diameter shell, inside, inches	15
Thickness, inch	7/16
Tube sheet	Wrought steel
Thickness, inches	1½
Diameter, inches	23
Tubes	Seamless drawn steel
Number of tubes	55
Diameter of tubes (outside), inch	¾
Thickness of tubes, inch	.065

Heating surface (inside surface), square feet	120
Diameter fuel oil inlet and outlet, inches	2½
Diameter steam connection, inches	1½
Diameter drain connection, inch	1
Retarders in first pass tubes	55
Number of turns in first pass retarders	3
Retarders in second pass tubes	49
Number of turns in second pass retarders	6

The oil heaters are fitted with gauge glasses, and the fuel oil lines with air chambers, pressure gauges and thermometers at points convenient for observation.

A small hand-operated fuel oil pump is provided for use when lighting off and for emergency use.

MAIN PROPELLING MACHINERY.

There are four (4) main propelling units, each consisting of a high pressure and a low pressure turbine, of the well known Parson's type, each in a separate casing, and driving its line shaft and propeller through reduction gearing.

An astern turbine of the two stage Cramp Impulse type is fitted in each low pressure casing, and a cruising turbine, also of the Cramp Impulse type, with hydraulic clutch and reduction gear is fitted to the forward end of the high pressure rotor on No. 1 and No. 3 shafts.

The rotor drums are wrought steel, shrunk and pinned to the shaft studs which form the ends.

The end thrust on the high pressure rotor is partially balanced by the contact balance dummy, while balance in the low pressure is maintained by dividing the steam flow.

Thrust bearings of the Kingsbury type are used on all turbine rotors to prevent end motion, as well as on the main reduction gears to take the propeller thrust.

Forced lubrication is used on all turbine and reduction gear bearings.

PARTICULARS OF HIGH PRESSURE TURBINE BLADING.

Length of Cylinder, feet and inches 7–7

Expansion No.....	1	2	3	4	5	6	7	8	9	10
Number rows blades.....	9	9	8	8	4	4	3	3	3	3
Height of blades, inches.....	$\frac{5}{8}$	$\frac{13}{16}$	1	$1\frac{3}{8}$	$1\frac{3}{8}$	2	$2\frac{1}{2}$	$3\frac{1}{8}$	$3\frac{7}{8}$	$3\frac{7}{8}$
Pitch of blades, inches.....	$1\frac{3}{16}$	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{9}{16}$	$1\frac{9}{16}$	$1\frac{13}{16}$	$1\frac{13}{16}$	$2\frac{1}{4}$	$2\frac{3}{16}$
Clearance, radial...	.03	.03	.03	.035	.035	.04	.04	.045	.045	.045
Approx. blades per row in cylinder..	354	360	364	374	274	286	228	236	210	206
Approx. blades per row in rotor.....	450	450	450	450	276	276	216	216	191	194

Diameter dummy drum, inches	29
Number rows dummy strips	20
Clearance, axial	.015
Oil float, axial	.012
Diameter labyrinth packing, inches	9½
Number of strips, inside	12
Number of strips, outside	5
Diameter of bearings, inches	7
Length of bearings, inches	7
Diameter of steam inlets:	
First cruising, 1st expansion, inches	8
Second cruising, 3rd expansion, inches	10
Main steam, 5th expansion, inches	13
Auxiliary exhaust steam, 7th expansion, inches	6
Diameter of exhaust to low pressure turbine, inches	25

PARTICULARS OF LOW PRESSURE TURBINE BLADING.

Length of Cylinders (2), feet and inches..... 2–11¼

Expansion Number ...	1	2	3	4	5	6	7	8	9	10	11
Number of rows of blades.....	2	2	2	2	2	2	1	1	1	1	1
Height of blades, inches.....	2⅛	2 1⅜	3¾	5	6⅝	8¾	10½	12	12	12	12
Pitch of blades, inches.....	1 5/16	1 5/16	1⅛	1⅛	1⅞	2 3/16	2¼	2⅛	2¼	2⅞	2 1⅛
Clearance radial.....	.065	.065	.070	.075	.080	.085	.090	.095	.095	.095	.095
Approx. No. blades per row in cylinder.....	634	650	510	534	434	396	418	374	374	374	336
Approx. No. blades per row in rotor.....	777	777	496	496	390	344	344	291	299	304	278

Diameter rotor drum, feet..... 4
Diameter labyrinth packing, inches..... 12
Number of strips, inside..... 12
Number of strips, outside..... 5
Diameter of bearings, inches..... 9½
Length of bearings, inches..... 13
Diameter of inlet from high pressure turbine, inches..... 25
Exhaust trunks to main condenser, two (2) each, feet and inches. 4–1 × 7

PARTICULARS OF ASTERN TURBINE BLADING.

Type, Cramp Impulse.
Number of stages..... 2

	First Stage					Second Stage		
	1	2	4	5	6	3	7	8
Number of blades in row, cylinder.....	394	400	...	...	...	418	...	...
Number of blades in row, rotor.....	...	...	363	357	354	...	341	333
Nozzle ring, number of ports			60				112	
Area, square inches.....			17.7				140	

Mean diameter center of blades, feet..... 5
Diameter astern steam inlets, 2 each, inches..... 7½

PARTICULARS OF CRUISING TURBINE BLADING.

Type, Cramp Impulse.
Number of stages..... 2

	First Stage			Second Stage		
	1	3	4	2	5	6
Number of blades in row, cylinder.....	208	...	...	208	...	...
Number of blades in row, rotor.....	...	191	188	...	191	188
Nozzle ring, number of ports.....		80			70	
Area, square inches		6.15			5.3812	

Mean diameter center of blades, inches..... 24
Diameter of main steam connection, inches..... 6½
Diameter second stage by-pass, inches..... 8
Diameter exhaust to high pressure turbine, inches..... 10
Diameter shaft bearings, inches..... 3
Length shaft bearings, inches..... 4¼

In the cruising turbine the nozzles in the first stage are arranged in groups and controlled by valves as follows:

	Diameter inches	Number of Nozzles
Throttle No. 0	6½	24
Valve No. 1	4	28
Valve No. 2	1½	4
Valve No. 3	2	8
Valve No. 4	3	16

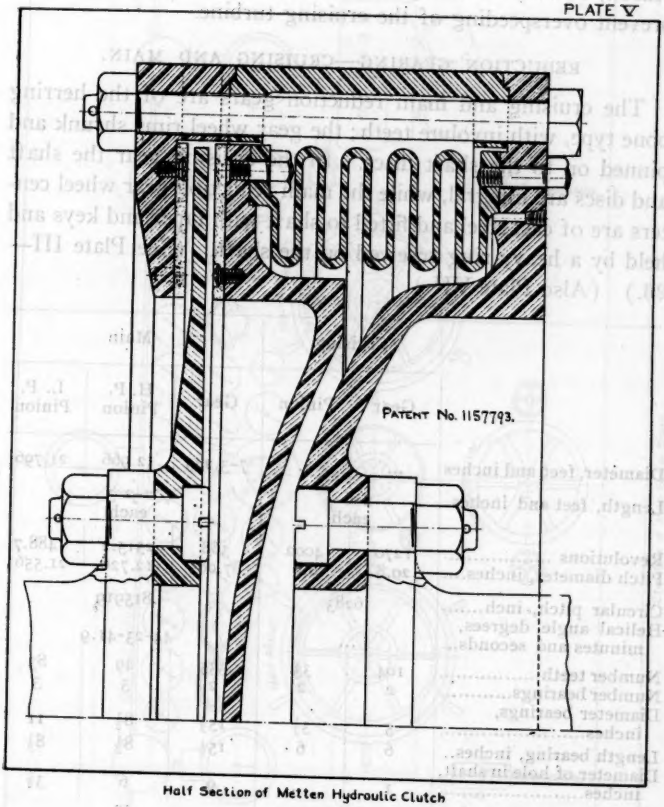
This allows for operating combinations whereby any power from 30 to 100 per cent may be had.

Nozzle Combination	Number of Nozzles Engaged	Per cent of Total
0	24	30
0 - 2	28	35
0 - 3	32	40
0 - 2 - 3	36	45
0 - 4	40	50
0 - 2 - 4	44	55
0 - 3 - 4	48	60
0 - 1	52	65
0 - 1 - 2	56	70
0 - 1 - 3	60	75
0 - 1 - 2 - 3	64	80
0 - 1 - 4	68	85
0 - 1 - 2 - 4	72	90
0 - 1 - 3 - 4	76	95
0 - 1 - 2 - 3 - 4	80	100

Each cruising turbine drives the high pressure rotor through a reduction gear and a Metten hydraulic clutch, oil pressure for operating which is supplied by a small gear pump driven from the cruising gear shaft. A steam operated pump is also provided for starting and emergency use.

Plate 5 shows a half section of the hydraulic clutch, oil for operating which enters through the hollow shaft of the cruising reduction gear. (See Plate III—27.)

PLATE V



Half Section of Metten Hydraulic Clutch

A balanced piston valve, hydraulically controlled by the clutch oil pressure, is placed in the steam line to cruising turbine, and so arranged that in case the clutch oil pressure fails for any reason, thereby releasing the clutch, this valve will close and prevent overspeeding of the cruising turbine.

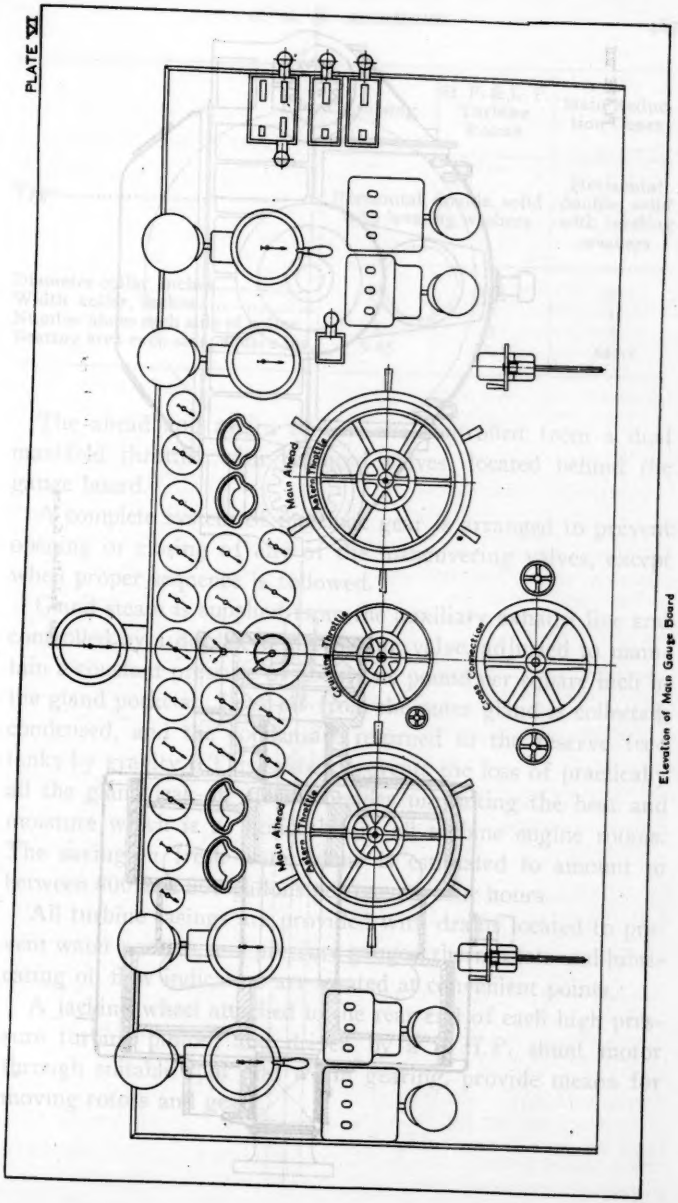
REDUCTION GEARING—CRUISING AND MAIN.

The cruising and main reduction gears are of the herring bone type, with involute teeth; the gear wheel rims shrunk and pinned on to the shaft discs. In the cruising gear the shaft and discs are integral, while the main reduction gear wheel centers are of cast steel and fitted to shaft with taper and keys and held by a heavy ring screwed on the shaft. (See Plate III—26.) (Also Plate VII.)

	Cruising		Main		
	Gear	Pinion	Gear	H. P. Pinion	L. P. Pinion
Diameter, feet and inches	...	...	7-3.245	12.966	21.796
Length, feet and inches..	2-6-½ each			2-2 3 each	
Revolutions	1270	4002	368	2515.2	1488.7
Pitch diameter, inches....	20.8	6.6	87.005	12.726	21.556
Circular pitch, inch.....	.6283			.815919	
Helical angle, degrees, minutes and seconds...	...			44-23-41.9	
Number teeth.....	104	33	335	49	83
Number bearings.....	2	2	2	3	3
Diameter bearings, inches.....	5	3½	15½	8½	11
Length bearing, inches..	6	6	15½	8½	8½
Diameter of hole in shaft, inches.....	1	...	9	6	3½
Number jaws in coupling	6			12	

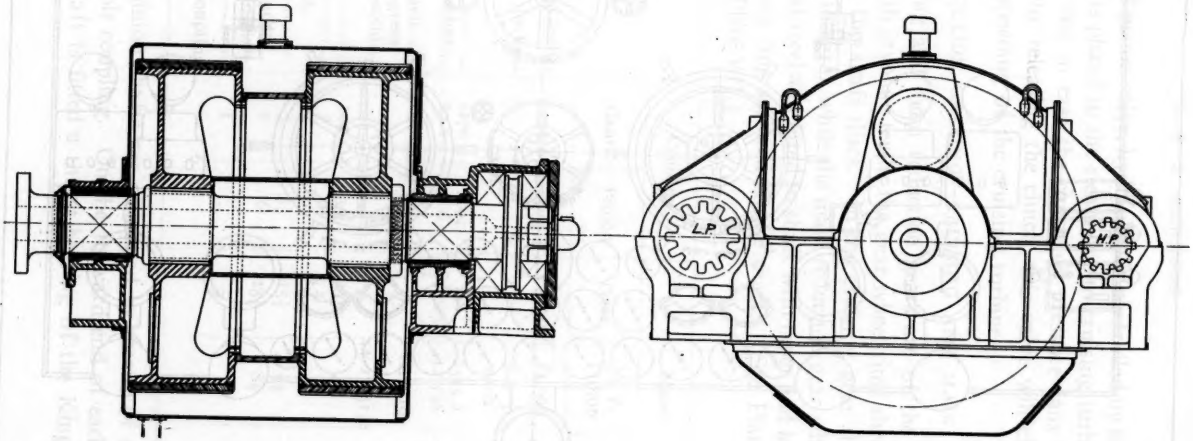
Rotor and pinion shafts are connected by means of a ring and claw flexible coupling. On the forward end of each gear and turbine shaft is fitted a thrust bearing of the Kingsbury type.

PLATE VI



Elevation of Main Gauge Board

PLATE XII



Main Reduction Gears

	Cruising	H. P. & L. P. Turbine Rotors	Main Reduction Gears
Type	Horizontal, double, solid with leveling washers		Horizontal double, solid with leveling washers
Diameter collar, inches	5½	13½	30
Width collar, inches	1	2	4½
Number shoes each side of collar	3	3	6
Bearing area each side, square ins.	6.45	51.1	448.2

The ahead and astern turbine are controlled from a dual manifold throttle, with balanced valves, located behind the gauge board.

A complete system of interlock gear is arranged to prevent opening or closing of any of the maneuvering valves, except when proper sequence is followed.

Gland steam is supplied from the auxiliary exhaust line and controlled by a dead weight reducing valve, adjusted to maintain a constant pressure of about one pound per square inch in the gland pockets. Leak-off from the outer gland is collected, condensed, and the condensate returned to the reserve feed tanks by gravity. This system prevents the loss of practically all the gland leak-off steam, besides preventing the heat and moisture which is so noticeable in all turbine engine rooms. The saving in fresh water alone is estimated to amount to between 600 and 800 gallons per twenty-four hours.

All turbine casings are provided with drains located to prevent water pockets, and pressure gauges, thermostats and lubricating oil flow indicators are located at convenient points.

A jacking wheel attached to the rear end of each high pressure turbine pinion, and driven by a 8 H.P. shunt motor through suitable spur and worm gearing, provide means for moving rotors and gears.

PARTICULARS OF CONDENSERS.

	Main	Augmentor	Auxiliary
SHELL:			
Material	Steel	Composition	Steel
Length, feet and inches	12–3	3–6	6–7
Diameter, feet and inches	7–3¾	21	2–11¾
Diameter exhaust trunks, feet and inches, each	(2) 5–9 5–1	...	...
Diameter auxiliary exhaust inlet, inches	10	...	11
Diameter inlet from Radojet, inches	...	8	...
Diameter air pump suction, inches	15	6	6
Diameter augmentor condenser suction, inches	8	...	...
HEADS:			
Material	Composition	Composition	Composition
Diameter circulating water inlet, inches	28	4	6
Diameter circulating water outlet, inches	28	4	6
Number passes, circulating water	1	2	2
TUBE SHEETS:			
Material	Composition	Composition	Composition
Diameter, feet and inches	7–9¼	2–⅞	3–4⅜
Thickness, inches	1¼	⅞	1
Length between, feet and inches	12–3	3–6	6–7
TUBES:			
Material	Composition	Composition	Composition
Number, one condenser	5625	306	930
Diameter, inches	⅝	⅝	⅝
Thickness, inch	.049	.049	.049
	Straight	Curved	Straight
Cooling Surface: square feet	11272	175	1001.63
S. H. P. per condenser	22500	...	...
Ratio C. S. to S. H. P.	.5	...	...
Radojet Air Pump, made by C. H. Wheeler Mfg. Co., Philadelphia, Pa.			
Diameter steam inlet, inches	...	2½	...
Diameter air inlet (2), inches	...	5	...
Diameter exhaust to augmentor condenser, inches	...	8	...
Designed steam consumption, pounds per hour	...	675	...
Pounds air compressed per hour at 80 degrees per F	...	220.	...

MAIN AND AUXILIARY CONDENSING SYSTEMS.

The maintenance of a high vacuum is so necessary to the economical operation of a steam turbine that particular care has been used in the layout on this vessel.

The main condensers, four (4) in number, are circular in section with steel shells and composition tube sheets and sea water chests. (See Plate III—22.)

The tubes are straight, one end expanded in the tube sheet and the other end packed with the conventional ferrule and packing.

Cooling water is supplied through scoops in the ship's bottom, and makes a single pass through the tubes. The flow of circulating water is assisted when warming up, running at low speeds or maneuvering, by a turbine driven centrifugal pump.

The overboard discharge from the forward engine room condensers is carried well outboard from the intake of the after engine room condensers to prevent their getting the warm discharge water.

The main condensers are each fitted with a high and low suction, the low suction leading to the main air pump, while the high suction leads to a Radojet air pump, (See Plate III—23) which in turn discharges into an augmentor condenser, from which suction is taken by the main air pump. The augmentor condenser has composition shell heads and tube sheets and tubes are curved and expanded into tube sheets at both ends. The auxiliary is similar in construction to the main condenser, *i.e.* steel shell, composition tube sheets, straight tubes expanded into tube sheet at one end and packed at the other. There is one auxiliary condenser in each engine room.

FORCED LUBRICATION SYSTEM.

The forced lubrication system consists of drain tanks located at the rear of each engine room, into which all main turbine, and reduction gear bearings drain. The lubricating oil pumps take suction from these drain tanks and force the oil through strainers, and oil coolers to the lines supplying the

bearings. Cross-connection lines provide for any necessary combination of pumps and coolers in one engine room, but each engine room has a complete and independent system. Centrifugal oil separators of the DeLaval type are provided in each engine room. (See Plate IV—24.)

The lubricating oil coolers, four (4) in each engine room, have steel shell, with straight tubes expanded into composition tube sheets at both ends, expansion being taken care of by bellows joint at center of cooler shell.

The oil is outside the tubes and baffles compel the oil to make eleven (11) passes over the tubes before reaching the outlet.

SHELLS:

Material	Steel
Length, feet	7
Diameter, feet and inches	2–1½
Diameter oil inlet, inches	5
Diameter oil outlet, inches	5

HEADS:

Material	Composition
Diameter cooling water inlet, inches	4½
Diameter cooling water outlet, inches	4½
Number of passes of water through tubes	2

TUBE SHEETS:

Material	Composition
Diameter, feet and inches	2–6
Length between, feet	7
Thickness, inches	1⅛

TUBES:

Material	Composition
Diameter, inch	⅝
Thickness, inch	.049
Number of tubes (1 cooler)	522
Cooling surface, square feet	597

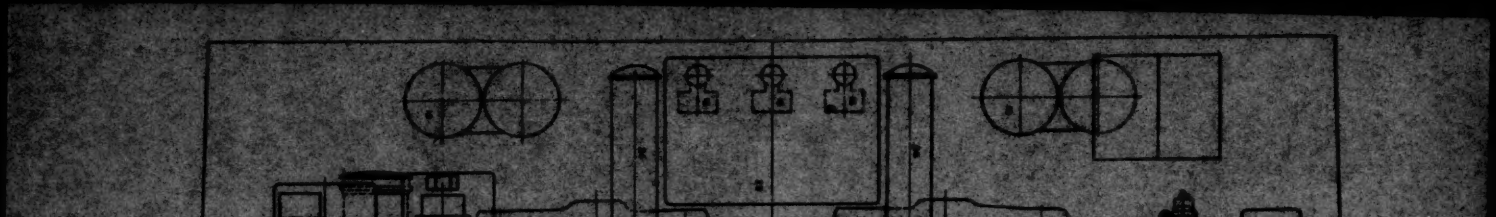


PLATE IX

Legend of After Machining Components

Each cooler is designed to maintain the lubricating oil supply to each combined turbine and gear unit for one shaft at a temperature not exceeding 100 degrees F. when supplied with circulating water at 70 degrees F.

MAIN FEED AND FILTER TANKS.

There is one 3674 gallon feed and filter tank located in after end of each engine room which takes the discharge from main and auxiliary air pumps, make-up feed pump, main steam line, heater and drain traps, and from which suction is taken by the main and auxiliary feed pumps. (See Plate III—25.)

SHAFTING AND PROPELLERS.

The main line shafting is of forged nickel steel, hollow the entire length and having flanged and bolted couplings, except the stern tube and propeller shaft lengths, which are fitted with special taper sleeve coupling to facilitate installation.

A Gary-Cummings torsion meter is fitted in each shaft, the recording tables all being located abreast and in the coupling just abaft the gear case in after engine room.

All shafts rake outboard and down and are carried on self aligning, ring oiled bearings. The entire lower half of each bearing is babbitted while in the upper half only a 2 5/8 inch strip at each end is babbitted. A small coil for cooling water is provided in the oil pocket, and a lubricating oil supply connection is fitted to each bearing.

PUMPS AND PUMP CONNECTIONS.

The following is a complete list of pumps as installed, giving type, capacity and use; the numbers following text in location column refer to plates illustrating this article, from which the relative location in each machinery compartment can be seen at a glance:

	Outboard	Inboard
Total length, feet and inches...	202-10 1/16	159-3 3/4
Number of section, exclusive of stern tube and propellers...	4	3
Diameter shafting, inches.....	14 1/2	14 1/2
Diameter axial hole, inches.....	9 1/8	9 1/8
Diameter bearings, inches.....	14 3/4	14 3/4
Length bearings, feet and inches	2-10	2-10
Number of bearings	7	5
Diameter coupling, flanges, feet and inches.....	2-2 1/2	2-2 1/2
Thickness of flanges, inches.....	3 1/8	3 1/8
Number of bolts per flange.....	8	8
Diameter bolts, tapered, inches..	3	3
Length of sections, maximum, feet and inches.....	32-5	28-3/4
Length of sections, minimum, feet and inches..	26-11 1/4	26-9 3/4
Stern tube shaft length, feet and inches.....	42-5/8	38-4 1/8
Propeller shaft length, feet and inches.....	38-9 5/8	38-9 5/8
Length coupling, feet and inches.....	4-4 1/2	4-4 1/2
Length forward stern tube bearing, feet.....	5	5
Length after stern tube bearing, feet and inches.....	6-3	6-3
Length strut bearing, feet and inches.....	6-3	6-3
Diameter strut bearing, inches..	17	17
Distance end of shaft from center line, feet and inches.....	21-1 7/16	10-6
Distance center propeller hub, up from boss line, feet and inches.....	3-5	3-5
Rake—per foot of length, down, inches.....	.164948	.421666
Rake—per foot of length, outboard inches.....	.239369	.179375
Number of propellers	2	2
Material propellers.....	Manganese bronze	Manganese bronze
Starboard propellers.....	Right hand	Right hand
Port propellers.....	Left hand	Left hand
Number of blades.....	3	3
Diameter, feet and inches.....	11-4	11-4
Pitch, feet.....	12	12
Expanded area, square feet.....	69.62	69.62
Projected area, square feet.....	61.32	61.32
Expanded area÷disc area.....	.691	
Projected area÷disc area.....	.609	
Pitch÷diameter	1.06	
Speed knots	34.5	
R. P. M.....	368.	
S. H. P. total four (4) shafts.....	90,000	

Pumps	No.	Size and Capacity, inches	Kind	Inches	Suction pipes from	Inches	Discharge pipe to	Location
Main air.....	4	Two 36 × 27	Twin beam single acting	16 6	Condensers through water seals Augmentor condensers	13	Feed tank	Two in each engine room III–13
Main condenser circulating....	4	10,500 gallons per minute	Centrifugal, turbine driven	19 × 27 7	Sea chest Bilge	19	Main condensers	Two in each engine room III-12
Main feed.....	4	600 gallons per minute	Three stage centrifugal turbine driven	6	Feed suction pipe	5	Main feed discharge	Two in each engine room III–14
Auxiliary feed.	4	15 × 10 × 18	Vertical, piston double acting simplex	6 4 1½	Feed suction line Reserve feed tanks Hose connection	5 1½	Auxiliary feed Hose connection	One in each boiler room II–6
Engine room fire and bilge.	4	10 × 9 × 12	Vertical, piston double acting simplex	4½ 4½ 1½	Sea Drainage Hose connection	4 4 4 1½	Fire main Overboard Flushing system through fire main Hose connection	Two in each engine room III–15
Fire room fire and bilge.....	2	10 × 9 × 12	Vertical, piston double acting simplex	5 5 1½	Sea Drainage Hose connection	4½ 4½ 1½	Fire main Overboard Hose connection	One in Number 2 and one in Number 3 boiler room II–7
Auxiliary condenser, air.....	2	7½ × 14 × 12	Vertical, double acting simplex	6	Auxiliary condenser	5	Feed tank	One in each engine room III–17

Pumps	No.	Size and capacity inches	Kind	Inches	Suction pipes from	Inches	Discharge pipe to	Location
Auxiliary condenser circulating and oil cooler.....	2	...	Centrifugal turbine driven	6	Sea	6	Auxiliary condenser	One in each engine room
						6	Oil coolers	III–18
Evaporator feed	2	4½ × 6 × 6	Vertical, piston double acting simplex	2½	Sea	2	Evaporators through feed heaters	In evaporator room
				2½	Distiller circulating water discharge			
Distiller fresh water.....	2	4½ × 5 × 6	Vertical, piston double acting simplex	2½	Distiller testing tanks	2	Reserve feed tanks	In evaporator room
				2½	Bottom of Distillers	2	Ship's tanks	
						2	Main feed tanks	
Distiller circulating.....	2	...	Centrifugal turbine driven	6	Sea	5	Distillers	In evaporator room
Distiller air.....	1	6½ × 10 × 8	Vertical, double acting simplex	4	Distillers	3	Through manifold to testing tank and feed water pump suction	In evaporator room
Evaporator brine.....	1	4½ × 6 × 6	Vertical, piston double acting simplex	2½	Evaporators	2	Overboard	In evaporator room
Lubricating oil..	7	8 × 12 × 12	Vertical, piston double acting simplex	6	Lubricating oil drain tanks	5	Bearings and reduction gears	Four in forward and three in after engine room
						2	Settling tank	III–16

Pumps	No.	Size and capacity, inches	Kind	Inches	Suction pipes from	Inches	Discharge pipe to	Location
Fuel oil service..	8	5½ × 4 × 8	Vertical, piston double acting simplex	4½ 4½	Oil storage tauk Booster pump discharge	2½	Burners	Two in each boiler compartment II–8
Fuel oil booster	2	8 × 10 × 12	Vertical, piston double acting simplex	6 6	Oil storage tanks Deck hose connection	4½ 5 5 1	Service pump supply Storage tanks Deck hose connection Galley range tank	One in No. 1 and one in No. 4 boiler compartment II–5
Oil cooler circulating.....	2	1,000 gals. per minute	Centrifugal turbine driven	6	Sea	6	Oil coolers	One in each engine room
Reserve feed water.....	2	7 × 7 × 12	Vertical, piston double acting simplex	4 4	Reserve feed tank C & R pipes from ship's side	3½ 3½	Main feed tank Tank suction line in F. E. R.	One in each engine room III–19
Clutch oil.....	2	3½ × 1 – 3/4 × 4	Vertical, piston double acting simplex	¾	Main turbine lubricating supply line	¾	Cruising turbine lubrication and clutch	One in each engine room III–20

R. P. M. STANDARDIZATION RUNS

Run	Knots	Revolutions per Minute — Shaft Numbers: 1	2	3	4	Shaft Horsepower — Shaft Numbers: 1	2	3	4	Total S. H. P.	Remarks
1	9	80.36	82.36	80.75	83.42	248	194	203	275	920	
2	9	82.82	82.63	81.94	81.89	261	202	222	269	954	
3	9	84.46	85.47	86.03	86.07	267	231	279	283	1,060	
4	12	109.21	112.31	111.32	109.39	571	668	702	615	2,556	
5	12	108.45	110.85	110.62	112.32	513	840	589	612	2,554	
6	12	110.16	109.64	109.52	108.87	584	673	672	593	2,522	
7	17	159.89	160.9	162.29	162.12	1,663	2,059	1,888	1,955	7,565	
8	17	161.88	162.91	162.62	162.42	1,725	2,322	2,054	1,983	8,084	
9	17	160.71	160.26	161.73	164.34	1,764	2,124	1,894	2,022	7,804	
10	22	209.03	205.66	206.43	207.55	4,062	4,617	4,205	4,060	16,944	
11	22	207.93	206.6	206.31	209.46	3,692	4,376	3,979	4,113	16,160	
12	22	207.75	201.63	203.16	205.69	3,955	4,172	3,957	3,988	16,072	
13	25	220.39	223.04	222.44	221.30	4,701	5,532	5,054	4,767	20,054	
14	25	223.26	223.03	226.27	223.66	5,076	5,731	5,692	5,095	21,594	
15	25	222.86	225.58	225.6	222.9	4,770	5,779	5,493	4,952	20,994	
16	27.5	261.78	258.95	260.56	260.65	8,302	8,476	8,246	8,325	33,349	
17	27.5	267.39	264.66	263.54	266.90	9,113	8,899	8,127	8,904	35,043	
18	27.5	264.84	258.44	258.76	264.22	8,588	8,368	7,769	8,917	33,642	
19	30	287.51	268.30	269.07	286.97	12,230	8,832	8,610	12,129	41,801	
20	30	299.23	293.80	296.15	298.75	13,445	12,557	12,841	12,849	51,692	
21	30	293.83	291.83	293.19	294.19	12,695	13,854	12,733	12,769	52,051	

R. P. M. STANDARDIZATION RUNS

Runs	Knots	Revolutions per Minute — Shaft Numbers: 1	2	3	4	Shaft Horsepower — Shaft Numbers: 1	2	3	4	Total S. H. P.	Remarks
22	32	312.59	308.79	309.28	309.83	13,505	13,364	13,327	13,632	53,828	
23	32	346.3	347.8	344.5	346.35	18,232	19,354	18,173	19,124	74,883	
24	32	326.9	329.5	326.8	327.2	17,210	18,540	17,527	16,553	69,830	
25	Full	...	...	...	...	...	...	..	...	...	Test run, no data.
26	Full	377.14	365.27	363 24	372.07	24,692	23,417	22,235	24,045	94,389	
27	"	374.75	366.03	362.0	360.35	24,138	23,594	22,326	23,777	93,835	
28	"	372.0	363.0	359.0	364.0	23,977	26,560	22,594	22,298	95,429	Data rejected
29	"	372.0	363.0	359.0	364.0	23,977	26,560	22,594	22,298	95,429	
30	"	...	...	...	...	...	...	...	...	...	
31	Full	354.3	345.0	343.0	352.0	22,089	20,810	20,139	22,022	85,060	Test run
32	Full	378.0	367.9	361.0	370.0	23,842	23,887	21,647	23,697	93,073	
33	"	375.0	364.0	360.0	367.0	24,152	23,642	22,396	23,722	93,912	
34	"	376.6	369.0	362.0	368.0	24,012	23,705	22,245	23,596	93,558	
35	"	373.9	364.8	359.7	364.4	24,175	23,743	22,308	23,163	93,389	
36	"	378.6	374.0	362.0	368.0	24,105	23,523	22,096	23,549	93,273	

STEAMING TRIALS.

Speed run	Full power	30 Knots	25 Knots	20 Knots cruising	20 Knots without cruising	15 Knots with cruising	15 Knots without cruising	10 Knots with cruising
Date	5/15/23	5/17/23	5/17/23	5/13/23	5/17/23	5/12/23	5/17/23	5/12/23
Duration, hours	4	4	6	4	2	4	2	4
R. P. M. mean	367.8	294.92	239.77	186.42	186.74	137.43	137.55	91.96
Knots corresponding	34.16	30.03	25.01	20.04	20.06	15.03	15.05	10.07
Shaft horsepower	92,802	50,528	26,106	11,665	11,774	4,635	4,661	1,391
No. of boilers in use	12	12	6	4	4	2	2	2
Average pressure at boilers,								
Fireroom No. 1	264.9	265.9	261.7	...	265.6	...	...	...
" No. 2	263.2	262.9	...	265.2	...	...	...	...
" No. 3	262.2	263.	...	...	...	...	...	...
" No. 4	263.7	265.4	263.8	265.9	265.4	265.	264.5	251.1
Average air pressure in F. R. in inches, water								
Fireroom No. 1	5.93	3.57	3.75	...	3.1	...	...	...
" No. 2	6.25	4.3	...	3.16	...	...	...	...
" No. 3	7.1	3.3	...	...	...	...	...	...
" No. 4	7.24	3.77	3.0	3.4	3.0	3.25	3.45	3.03
Pressure at turbine throttle forward engine room	261.3	266.6	257.5	265.	256.	259.7	261.	251.4
Pressure at turbine throttle after engine room	240.5	258.9	252.9	257.1	251.3	256.	251.3	245.6
Vacuum, Absolute pressure, inches Hg. forward engine Room:								
Starboard	2.	1.3	.9	.73	1.5	1.24	1.7	1.2
Port	1.9	1.2	1.1	.73	1.74	1.15	1.9	1.15
After engine room:								
Starboard	1.25	1.013	.941	1.05	1.39	1.56	1.59	1.66
Port	1.5	1.24	1.03	.903	1.86	1.20	1.925	1.39
Temperature injection water. Degrees F.								
Forward E. R.	42.5	44.2	44.7	42.	49.	42.5	51.4	41.3
After E. R.	43.8	45.1	45.1	42.8	49.5	43.9	52.3	42.6

STEAMING TRIALS.

The vessel left the Contractor's works early on May 9, 1923, and proceeded to Rockland, Maine, where the standardization trials were run under the supervision of the Board of Inspection and Survey.

Rockland, Maine, anchorage was used as a base for fueling and preparation for the steaming trials that followed the standardization runs.

Fog delayed the progress of the trials, which were not concluded until the morning of 19 May 1923, after which the vessel returned to the Contractor's yard.

The mean trial displacement during all trials was 7252 tons.

U. S. S. *LANGLEY*, OUR FIRST AIRCRAFT CARRIER.

By Lieut.-Commander H. R. Keller, U. S. N.,
Member.

CONCEPTION.

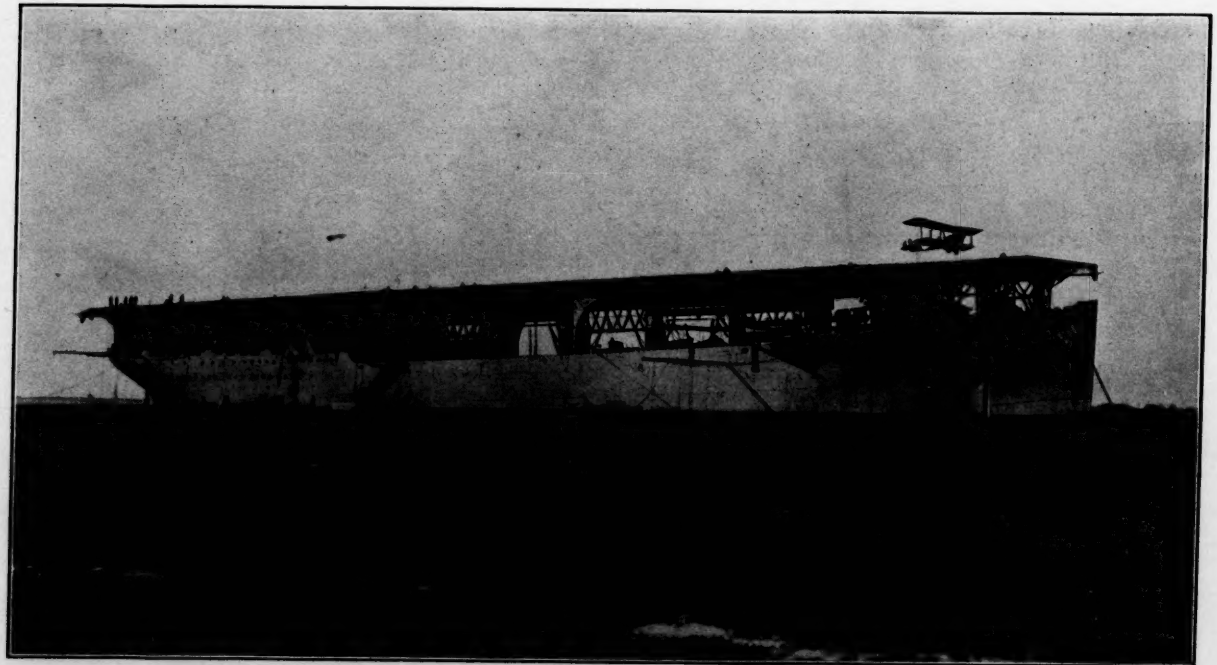
The U. S. S. *Langley*, our first aircraft carrier, owes its existence first to the General Board of the Navy who as early as 1915 and thereafter recommended such a type of ship to the Secretary of the Navy. But it is due more directly to the perseverance of several of our naval aviators, who during the war foresaw the need of such a type of ship, that the authority to convert the former collier *Jupiter* was obtained. Many other ships would have been more suitable, but the *Jupiter* seemed to possess the features necessary for an experimental aircraft carrier.

NAME.

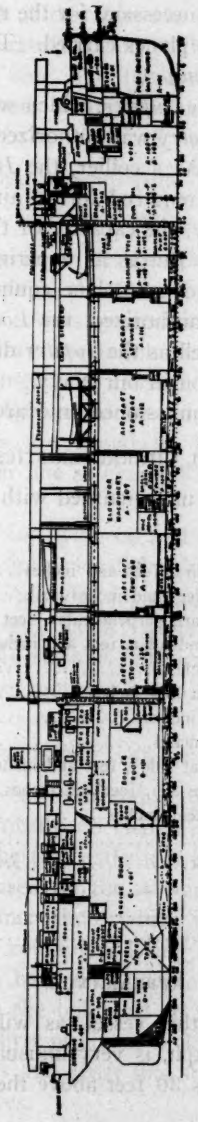
The *Langley* is named in memory of Professor Samuel Pierpont Langley, who was born in Roxbury, near Boston, Massachusetts, in August, 1834, and died in Aiken, South Carolina, February 27, 1906. He was a distinguished astronomer and physicist who received degrees from many universities and colleges for his eminent work in the field of astronomy as well as in the development of aviation. He was at one time an assistant professor of mathematics, United States Naval Academy, but his most notable work was in the field of aeronautics, in which he conducted many experiments in mechanical flight.

HISTORY AND MISSION.

It is the function of an aircraft carrier to provide a floating "landing field" for land as well as for seaplanes to take off from and to return to. Furthermore, an airplane carrier must



U. S. S. "LANGLEY."—OUR NAVY'S FIRST AIRCRAFT CARRIER.



INBOARD PROFILE.

provide all the facilities necessary for the repairs, maintenance, and operation of all the planes carried. These functions have been provided in the *Langley.*

The conversion of the *Jupiter,* which was our first electric-drive ship, into the *Langley* was authorized by act of Congress dated July 11, 1919. As a collier, the *Jupiter* had quite the opposite characteristics required for an aircraft carrier except for one feature; namely, the capacity of the former holds for the stowage of planes. But as an experimental ship, in preparation for the design of aeroplane equipment on board our large carriers recently authorized, the *Langley* is destined to fulfill her mission as well as the *Jupiter* did in revolutionizing the method of propulsion in our Navy.

The *Langley* was commissioned on March 20, 1922.

HULL CHARACTERISTICS.

For those who were unacquainted with the *Jupiter* the following data are given:

Length between perpendiculars, feet and inches	520–00
Length on load water line, feet and inches	520–00
Projection forward of forward perpendicular, feet and inches	2–00
Overhang aft of after perpendicular, feet and inches	20–2½
Length over all, feet and inches	542–2½
Length of straight keel, feet and inches	466–00
Breadth moulded, feet and inches	65–00
Breadth extreme, feet and inches	65–2 13/16
Depth moulded (main deck at side), feet and inches	39–9½
Depth as per rule (at Frame 86), feet and inches	39–6
Base line above bottom of keel, inches	1.6625
Ratio of length to beam	8.00
Coefficient of fineness block	.7215
Midship section	.9783
L. W. L. plane	.7954
Cylindrical	.7375

APPEARANCE.

The appearance of this vessel, as will be seen from the accompanying photograph, is very unusual. The flying deck, superimposed by towers 30 feet above the main deck, and the

smoke stack, set off at the side, are the most striking peculiarities from a moderate distance, while the various cranes, the radio booms at the side, and the aeroplanes on deck, when observed from a closer position, make the appearance more unique.

GENERAL DESCRIPTION OF HULL.

FLYING DECK.

This deck extends from Frame 5 to the stern and is the width of the ship, making the dimensions 65 feet by 523 feet. It is supported by 26 towers symmetrically located on opposite sides. These towers are rectangular in cross section and hollow, being made of angle bars crossed.

Access is obtained by two hatches on opposite sides at Frame 156. These hatches are closed by balanced hatch covers. Forward on the center line at Frame 32 a signal hatch affords access also.

The flying deck is for flying on and off and is covered with wood with expansion joints located at intervals. It contains the following facilities: Palisades for wind break; two anemometer masts; life nets; signal hatch; two telescopic masts; elevator hatch; gasoline supply stations; lubricating oil stations; two catapult tracks; radio compass house opening and arresting gear.

FORECASTLE AND UNDER.

The housed-in forecastle extends from Frame 33 forward. There has been practically no change in the forecastle since conversion. Below the forecastle deck is the paint room, lamp room, which is also used for the stowage of "dope" for aeroplane wings, and joiner shop, which is also a living space. The next lower deck contains a living compartment forward, in which a space has been set aside for the ship's tailor and sailmaker; and abaft the bulkhead at Frame 18 the aviation machine shop. The first platform deck contains a space used for boatswain's stores and abaft this store rooms for various ship and aviation stores.

The fore peak tank of 76,000 gallons capacity is below the first platform and extends to the chain locker at Frame 12. Abaft the chain locker is the pump room, containing a small drainage pump for pumping the forward compartments. An internal draft well is built into this compartment.

WELL DECK.

Just abaft the forecastle the well deck, which is a continuation of the main deck, extends to the break of the poop at Frame 139. An extension of the forecastle onto the well deck provides space for the assembly and repair officer's office on the starboard side of the well deck and a printing shop on the port side. The main deck contains the supports for the towers, operating stations for the elevator and both jib cranes; three gasoline supply stations, three lubricating oil supply stations, two motor boat platform cradles, stowage for one 24-foot motor sailing launch, and various holds for aeroplane stowage and mechanism for various machinery. There are 12 top-side ballast tanks located under the passageways outboard in the well deck.

POOP DECK AND UNDER.

The poop deck is a raised deck at the stern from Frame 139 aft. Above this deck is located an aerological laboratory on top of the flight officer's quarters which are built above the wardroom. Forward of the flight officer's quarters on the first deck is the sick bay, and forward of this, though separated by the athwartship stack is the intelligence office. This is also the senior flight officer's office. The wardroom extends from the cabin on the poop deck to the galley spaces. The latter contains the officers' and crew's galleys, the bake shop, the butcher shop, and the scullery. Between the interior passageways leading to the galley, there are offices for the navigator, first lieutenant, ordnance officer, disbursing officer, and supply officer. Forward of the galley spaces are the executive office, radio room, and radio emergency battery room.

Below the poop on the level of the main deck are the chief petty officer quarters, firemen's living quarters, general crew space, containing, in addition, canteen, post office, log room and chief petty officers' room, two crew's wash rooms, and steering engine room. This is the last complete deck aft. The flat forward of the engine room contains spaces for small store-issuing room, armory, radio repair room, instrument room, photographic laboratory, electrical workshop, brig, foundry and laundry. Abaft the engine room on the same level are situated the warrant officers' quarters, and at the stern is a space for crew's quarters. On the flat below this space are storerooms for wardroom, cabin, C. P. O., navigator, and medical department. Between these storerooms and what was formerly the after peak tank, a flat has been built for an after 5-inch magazine.

HOLDS AND INNER BOTTOMS.

Extending from the main deck to the inner bottoms just abaft the forecastle are two void spaces, separated by a fore and aft bulkhead. These voids separate two forward deep oil tanks from the two gasoline tanks. A cofferdam filled with water separates the gasoline tanks from the airplane stowage holds. Then, in order, are two aircraft holds, an elevator machinery compartment, and two additional aircraft holds, the latter equipped particularly for the stowage of aeroplane wings, propellers, lumber and miscellaneous gear. Below all aircraft holds are double bottoms containing fuel oil; below the gas tanks, cofferdam, and void forward of gas tanks are double bottoms which contain water.

ELEVATOR COMPARTMENT.

The elevator compartment is the space between Frames 86 and 104 which contains elevator machinery, gyro compass house, accumulators for air compressors, air compressor, and lubricating oil pumps. Hatches lead below this deck to a torpedo room containing torpedo storage racks, warhead locker,

5-inch magazine containing shell room, powder room and handling room, small arms magazine, bomb room, and four storerooms used for miscellaneous ship's stores. The elevator, when raised, is level with the flying deck. When lowered, it rests on the hatch above the main deck.

BOATS.

The following boats are supplied:

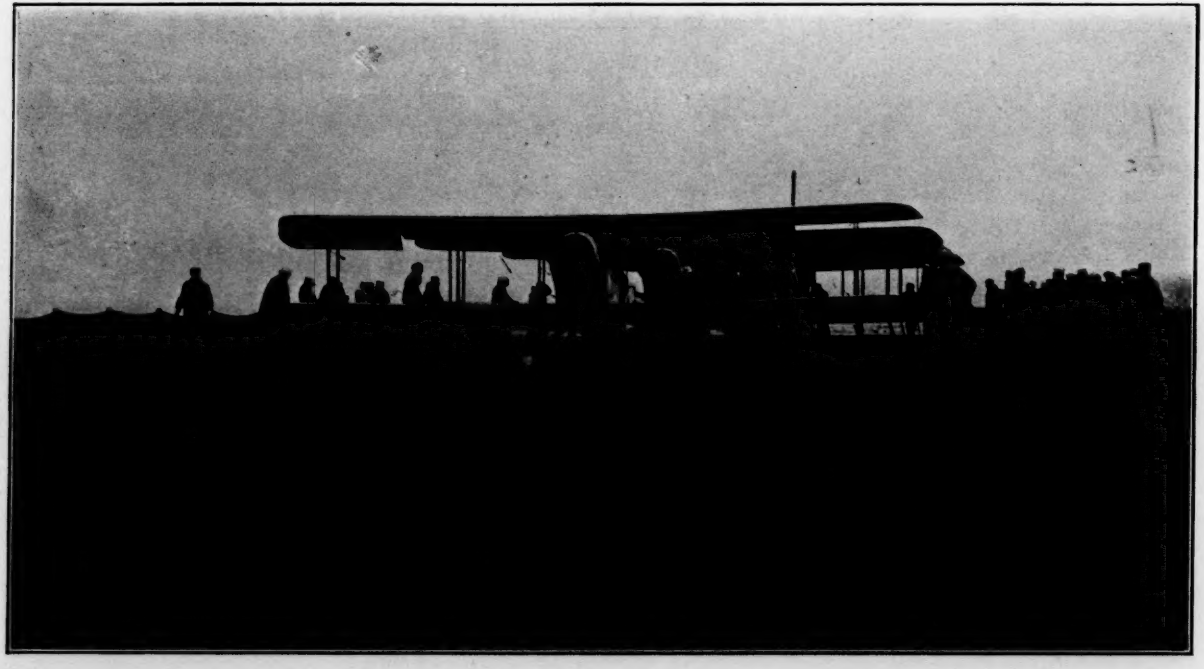
2—35-foot motor boats.
1—36-foot motor sailing launch, equipped with motor-driven winches for salvaging work.
1—36-foot motor sailing launch.
2—30-foot motor sailing launches.
1—24-foot motor sailing launch.
1—21-foot motor dory.
2—24-foot whaleboats.
2—20-foot dinghies.
2—14-foot wherries.
2—20-foot punts.

The motor boats, 24-foot motor sailers, dory and two whaleboats are stowed on the main deck, the whaleboats being swung outboard on their davits. All other boats are nested outboard on the same deck level as the flight officers' quarters.

LAUNDRY.

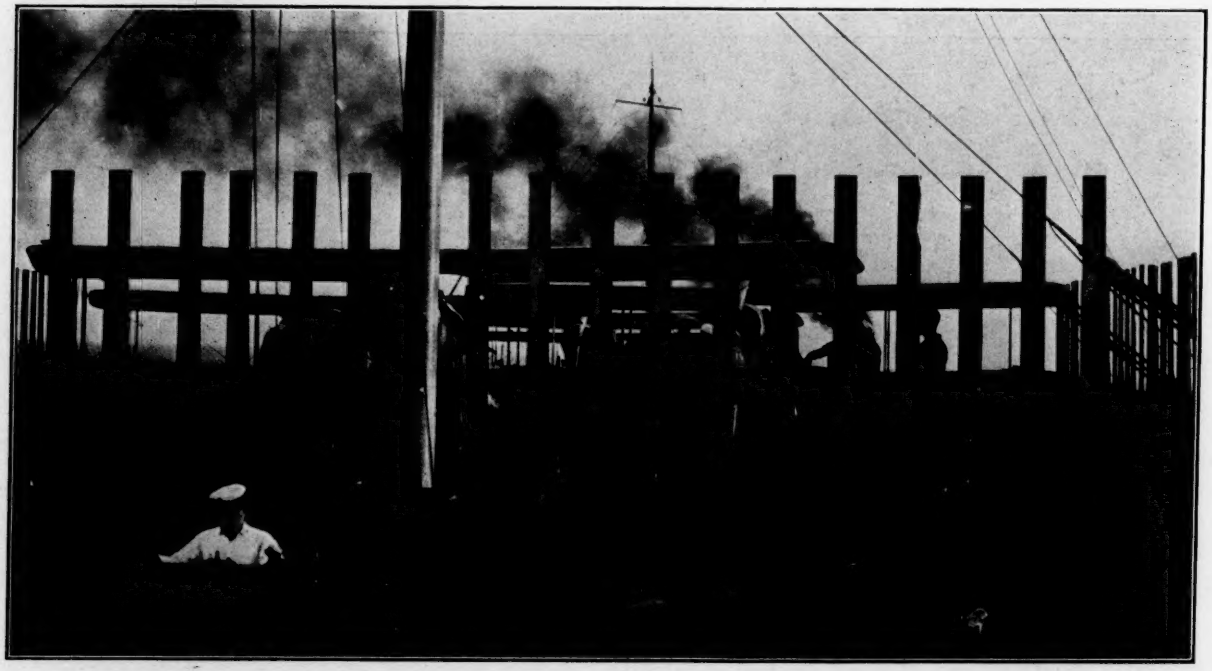
The laundry contains the following equipment:

1 Washer, 42 × 54 inches.
1 Extractor, 30 inches.
1 Tumble drier, 32 × 48 inches.
1 Flat work ironer, 48 inches.
1 Universal press, 38 inches.
1 Ironing table, 5 feet.
1 Stationary wash tub, 2 compartments.
1 Starch cooker.
1 Dry room, 42 × 56 inches.
3 Truck tubs, 22 × 32 inches.
1 Collar shaper.



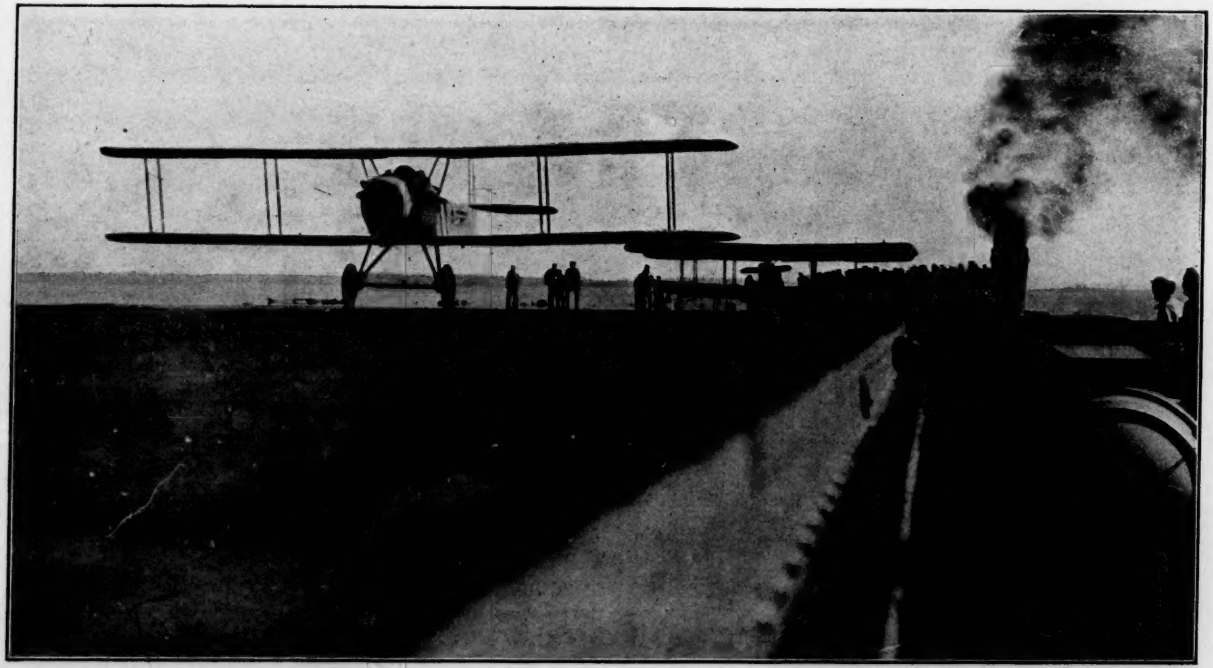
VIEW OF FLYING DECK SHOWING,

LIFE NETS. ACCESS HATCH.

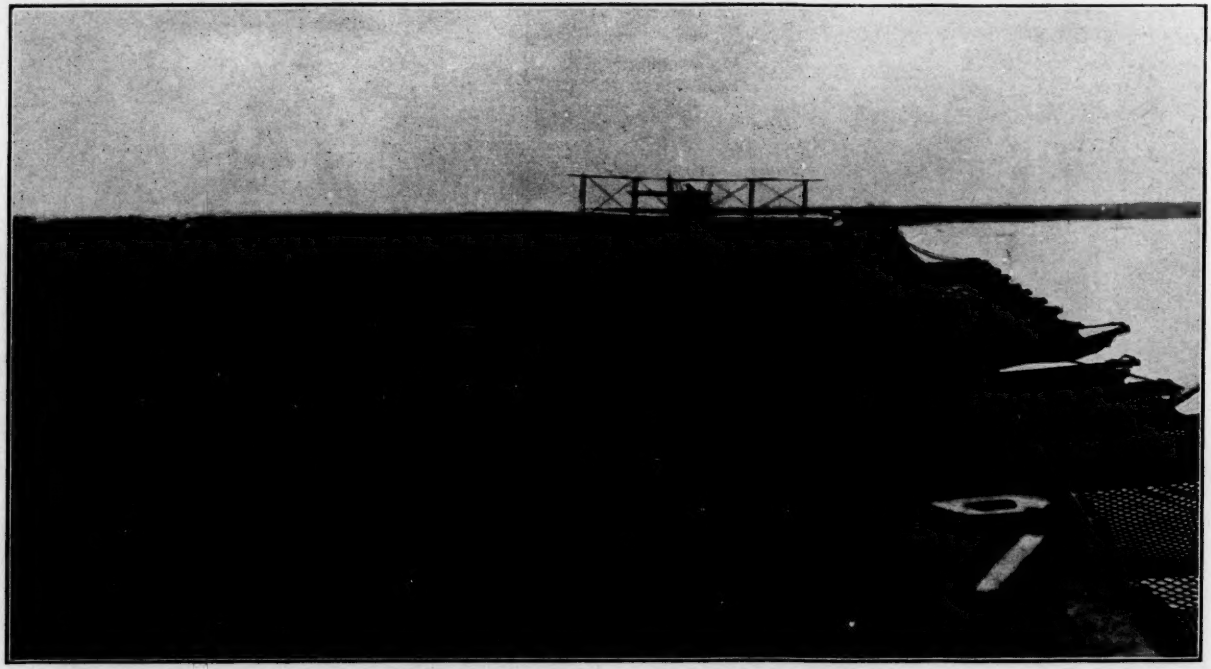


COMPASS HATCH.

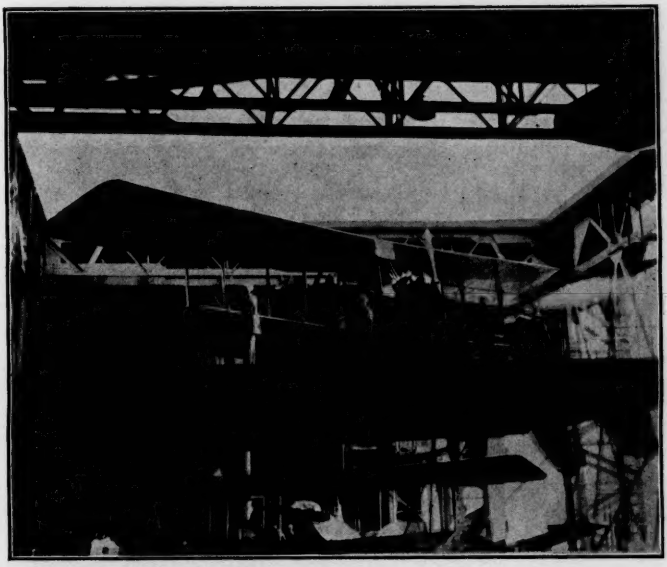
PALISADES.



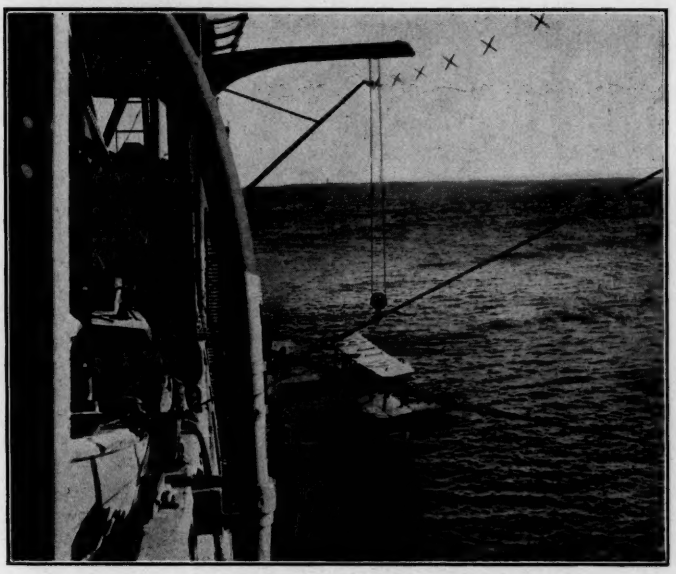
FLYING DECK SHOWING PLANES ABOUT TO TAKE OFF.



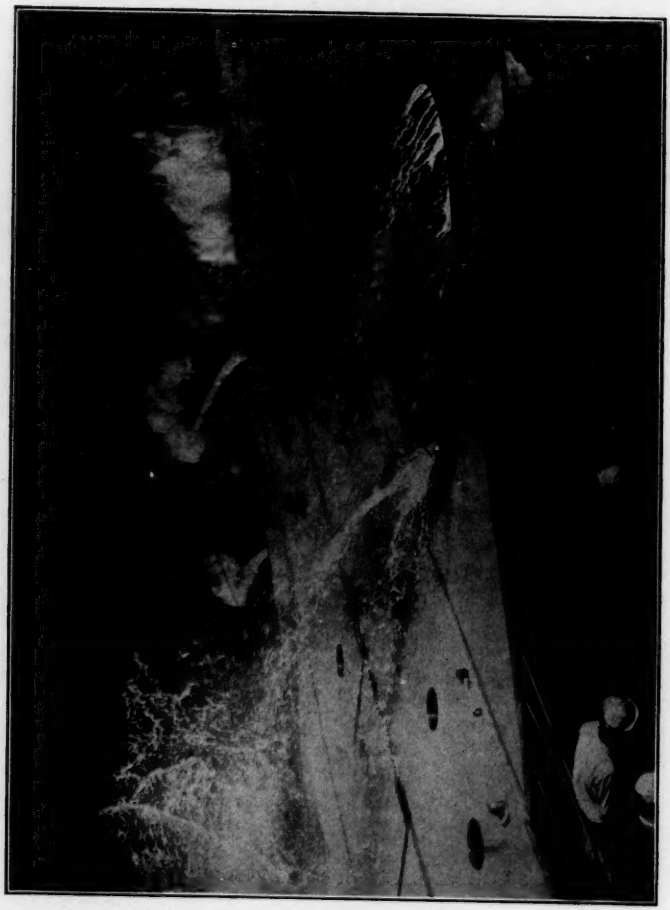
SEAPLANE BEING LAUNCHED FROM CATAPULT.



PLANE BEING LOWERED ON ELEVATOR.



JIB CRANE LIFTING SEAPLANE ON BOARD.
ALSO SHOWS RADIO BOOM WITH ANTENNA RIGGED.



VIEW SHOWING STARBOARD STACK AND SPRAY.

COMMISSARY FACILITIES.

There are three oil-burning Navy Standard ranges in the officers' galley and three in the crew's galley. Steam atomization is used normally, but the lines can be connected for air with little difficulty. The crew's galley, which is on the starboard side of an interior passageway on the poop deck leading from the crew's quarters, contains:

4 Steam-jacketed copper kettles, 40 gallons each.
1 Coffee urn, 60 gallons.
1 Kitchen and cake mixer, electric driven, 3 speeds.

Opposite the crew's galley is the butcher shop containing meat block and meat grinder and just aft of the butcher shop is the bake shop which has:

1—6-shelf electric bake oven, 120 volt, 140 ampère.
1—4-shelf standard navy proofer, No. 2.
1—Dough mixer, driven by 1 H.P. electric motor.
1—Dough trough.

Aft of the officers' galley is the vegetable room, which contains one potato peeling machine, electric driven, and four vegetable tubs. Opposite this room on the starboard side of the passageway is the scullery which contains shelves and racks for all mess gear and one electric driven dish washing machine.

The galleys are ventilated by exhaust blowers, but in spite of this the ventilation is inadequate.

SICK BAY.

The sick bay is very well equipped for any cases that it may be required to handle by a well lighted and equipped operating room. In addition there is an examining room, a consultation room, a ward with 8 beds, a bath room and an isolation ward. The sick bay forms a continuation of the flight officers' quarters forward of the latter.

PRINT SHOP.

The print shop is located on the port side of the well deck just aft of the break of the forecastle and contains the following equipment:

1 Printing press, 8 × 12 inches.
1 Paper cutter, 22 × 36 inches.
1 Stapler.
1 Stone.
1 Cabinet, type.

CAPSTAN AND WINCHES

The anchor engine is the same as was in use on the *Jupiter*. It is steam driven, contains two wildcats and two gypsy heads. The wildcats handle 2 11/16-inch chain.

There are four other steam driven hoisting winches located as follows:

1—After end of forecastle.
1—Just forward of break of poop.
1—Port side of after superstructure deck, at Frame 155.
1—Poop deck aft.

DRAINAGE.

The drainage facilities may be described under three headings; *i.e.*, forward, amidship and aft. By the forward system is meant all those compartments except fuel oil tanks forward of the first aircraft stowage hold. This includes the fore peak tank, void spaces forward of deep oil tanks, chain lockers, and void spaces on all sides of gasoline tanks. These compartments are pumped by a vertical duplex, 5¼ × 4 × 8-inch pump in a pump room just abaft the chain locker. Two-inch suction lines lead to each compartment except to the peak tank which has a three-inch suction.

The amidship system comprises all aircraft stowage holds together with all compartments below the elevator flat. Drain-

age of these is provided by a fire and bilge, or by a ballast pump (7½ × 9 × 10) in the engine room. A six-inch main leads forward from the pump in the engine room to the fire room where it branches and leads to manifolds on each side of the fire room, forward and outboard. Thence three-inch lines lead to each of the compartments drained by three manifolds.

The after system includes the fire rooms, engine room and all spaces abaft these. The same fire and bilge or ballast pumps are used on this system as for the amidship system. From the six-inch main above mentioned a branch is taken to a manifold on the after bulkhead of the fire room whence two three-inch suction lines lead to wells on each side of the fire room. A four-inch main leads aft to a manifold in the after part of the engine room and from this manifold three-inch branches lead to the after bilge wells in the engine room. Another four-inch branch from the main leads to a manifold near the pump. From this manifold one pipe leads to a two-inch bilge well near the pump and another to a similar well on the port side. Two-inch branches from the latter lead to the two spaces below the motors. Since what was formerly the after trimming tank is now used for a magazine, the four-inch suction lines leading to this space are now blanked off.

JOINER SHOP.

The joiner shop equipped for the combined use of ship's personnel and aviation personnel is equipped with:

No.	Machine	Horse-power
1	Saw, table, 36 × 42 inches, motor driven	5
1	Jointer, 12 × 60 inches, motor driven	3
1	Band saw, 30 inches, motor driven	2
1	Drill press, ½ inch, motor driven	1
1	Wood lathe, motor driven	3
1	Grinder, motor driven	
1	Saw-filing machine, motor driven	½
1	Bench planer	¼
1	Bench grinder	1

AVIATION FEATURES.

PALISADES.

The palisades are for the purpose of reducing the wind velocity within a limited area on the flying deck in order that planes may be overhauled or secured on that deck. They are located on the flying deck in the shape of a rectangle open at the after end. They are formed of five groups of channel bars about 6 inches wide by 12 feet long which are raised and lowered by five electric motors, two for each side and one for the front. The motors are 3 H.P. shunt wound.

There are 40 channels in the forward side of the rectangle and 20 in two groups on each side.

LIFE NETS.

Life nets extend on each side of the flying deck from stem to stern at a height of three feet above the deck. They are attached at intervals to bars which are hinged near the lower ends to permit raising vertically or lowering horizontally to a level about four feet below the height of the flying deck. These are for the "ground" personnel to stand in below the deck while planes are landing on the deck.

MASTS.

There are two telescopic masts, which extend about 50 feet above the flying deck when raised. One is at frame 39 and the other at frame 137. These masts raise and lower by a double fall, the standing part secured to the lower mast and the other end led to one of the deck winches. They step on a fid passing through the lower mast near the top. Wells 12 feet wide with flush covers provide a stowage place for the mast rigging.

CATAPULTS.

There are two catapults designed and manufactured by the Naval Aircraft Factory at Philadelphia. The tracks are 145 feet long for the after catapult and 180 feet for the forward

catapult. The former extends from the apron at the stern and the latter extends to the bow. Impulse is obtained by air, and a braking effect by pressure is obtained at the end of the travel of the "plough" which is connected to the carriage bearing the plane.

ELEVATOR.

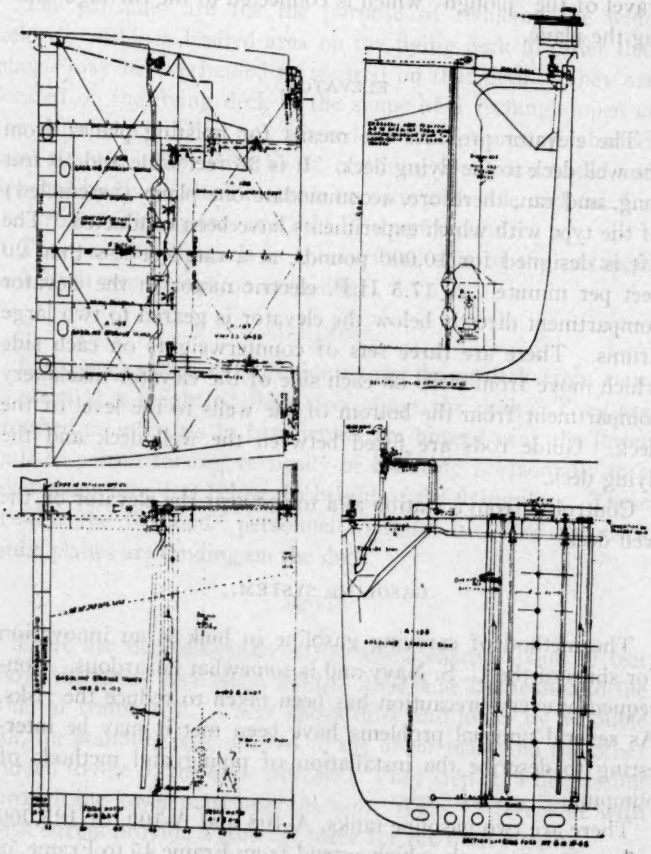
The elevator provides the means for hoisting planes from the well deck to the flying deck. It is 36 feet wide and 46 feet long, and can, therefore, accommodate one plane (assembled) of the type with which experiments have been conducted. The lift is designed for 10,000 pounds, at a rate not less than 20 feet per minute. A 17.5 H.P. electric motor in the elevator compartment directly below the elevator is geared to two large drums. There are three sets of counterweights on each side which move from wells on each side of the elevator machinery compartment from the bottom of the wells to the level of the deck. Guide rods are fitted between the well deck and the flying deck.

Control is from a station at a tower near the elevator on the well deck.

GASOLINE SYSTEM.

The method of carrying gasoline in bulk is an innovation for ships of the U. S. Navy and is somewhat hazardous. Consequently every precaution has been taken to reduce the risks. As several unusual problems have been met it may be interesting to describe the installation of piping and methods of pumping.

There are two gasoline tanks, A-108 and A-109, of 125,000 gallons capacity each, which extend from Frame 45 to Frame 52 between the inner bottoms and the main deck. Forward of these tanks and extending from Frame 38 to 45 there are two void spaces separated by a fore and aft bulkhead. In the port one of these the gasoline pump room is located. The pump, a



PIPING, GASOLINE SYSTEM.

vertical duplex 5¼ × 5 × 5, is in a water-tight room. Aft of the gasoline tanks a cofferdam, A-110, of 37,000 gallons capacity filled with fresh water forms an insulation between the gas tanks and the forward airplane stowage hold. The inner bottoms below the gasoline tanks as well as those below the void spaces and below the cofferdam are filled with fresh water. A small amount of fresh water is also kept on the deck of the void spaces to avoid the possibility of sparks igniting any gas that might result from leakage.

Ventilation for the void spaces to safeguard against any explosive mixture caused by leakage through the bulkheads or piping is provided by a motor inclosed in a water-tight compartment near the after end of the forecastle. A separate enclosed motor provides also for ventilation for the pump room. The gasoline tanks are inter-connected near the top by a 9-inch vent pipe (shown on the sketch). The pressure in the gasoline tanks may be kept at any determined amount, usually about one ounce, by the vent piping which leads from the top of each tank through six-inch piping above the main deck, through a 6-inch horizontal swing check valve and thence overboard by either one of the two methods. Normally, after passing the check valve which opens toward the outboard side of the ship, the gases flow upward through a return bend about ten feet above the main deck near the break of the forecastle and then pass through a wire screen into the atmosphere. When taking gasoline on board, or whenever for any reason it is desirable to accelerate the discharge of vapor, the connection to the return bend above mentioned is closed and the gases are directed downward through a pipe which leads through the adjacent top side tank. A one-inch connection from the forward flushing system induces a suction in this line. Self-recording temperature and pressure gauges are located on deck to show the exact condition of the tanks.

The tanks are filled by six-inch lines which lead from flanges on each side of the ship near the tanks across the deck. A six-inch outboard valve is provided in addition to individual six-

inch valves to each tank. The height of gasoline in the tanks may be equalized by means of the suction lines to the pump which may be inter-connected.

Gasoline may be taken from any level in either tank by opening the valve for that level from the deck connection and closing the valves to other levels. The pump discharge piping is so arranged above the deck that gasoline may be taken from one level and discharged at another level in order to circulate the gasoline and cool that nearer the top of the tank. There are four separate suction valves at different levels in each tank, each operated from the main deck.

Gasoline supply stations are located at convenient intervals on the port side of the main deck, poop and flying deck. Those on the main deck are at Frames 54 and 120. A station is provided at the after end of the poop deck for fueling seaplanes. The supply stations on the flying deck are beneath flush deck hatches at Frames 54, 69 and 137. Each station on the main deck has a two-inch connection as well as a three-quarter inch hose connection. Beneath the hatches on flying-off deck there is enough space in the compartment to coil down a gasoline hose. Cutout valves are so situated in the supply lines to the different stations that pressure is maintained at only a limited number of points when the pump is discharging to other supply stations.

The ship's regulations prevent smoking on the forecastle and well deck between tower No. 2, which is on the forecastle, and tower No. 8, which is located at Frame 120. "No smoking" signs are hung at frequent intervals. Steam smothering valves lead into each tank. It is felt that every precaution has been taken to make this installation as safe as possible.

LUBRICATING OIL.

Lubricating oil is carried in four tanks below the level of the elevator machinery deck. Two of these tanks are of 7,400 gallons capacity each, and the other two 8,400 gallons each. There are two 5 H.P. motor driven pumps on the elevator

machinery deck which take suctions from any of these tanks and discharge into any of the lubricating oil supply stations. There are three of these stations on the main deck at Frames 53, 70 and 120 and the filling station is located at Frame 88. On the flying deck there are three supply stations opposite the deck from the gasoline supply stations. A tank with steam coil is adjacent to the pump in the elevator space for heating the oil before discharging it to the supply stations.

TRAVELLING CRANES.

Two five-ton travelling cranes, one travelling from Frame 45 (just abaft the break of the forecastle) to the elevator hatch and the other from the elevator hatch to the break of the poop are necessary for the handling of planes and other weights. Control houses travel with the cranes.

The motive power and speeds for these cranes is supplied as shown.

Travel	Power	Speed
Bridge	15 K. W. motor, shunt wound,	200 feet per minute
Hoist	15 K. W. motor, shunt wound,	30 feet per minute
Trolley	7.5 K. W. motor, shunt wound,	100 feet per minute

JIB CRANES.

The two jib cranes provide a means of lifting seaplanes from the water to the deck as well for lifting land planes and other heavy weights from a dock. The forward one is on the starboard side of the well deck forward of the elevator and the after one on the port side of the well deck abaft the elevator. Two hinged girders, seventy-nine feet long, the forward one on the starboard side and the after one on the port side permit the jib cranes to operate without interference, while allowing them a maximum travel through an arc of 145 degrees, fifty degrees over the ship and ninety-five degrees over the side. The cranes are designed to rotate at a rate of 360 degrees per minute and hoist at 60 feet per minute.

The mechanism for operating these cranes consists of a 30 H.P. electric motor driving three Waterbury Hydraulic "A" motors which are connected to the "B" ends. These motors are located in the former top side tanks below the main deck. A separate mechanical control for hoisting, trolleying or swinging is provided in the tower adjacent to the crane.

PHOTO LABORATORY.

The photographic laboratory is an interesting part of the equpiment of this ship as it contains probably the most completely equipped office for both stationary and moving pictures on any ship now afloat. It is designed especially for use in connection with taking pictures of the planes taking off and landing on the decks in order to see the details of that work. For that purpose ultra rapid moving pictures are taken. These are reduced to very slow speed when exhibited so that the most minute and rapid changes of the planes are shown. Thus all irregularities in landing and taking off can be detected and corrected. Besides providing the equipment for taking, developing and exhibiting any type of stationary or moving picture, stationary photographs may be reproduced or enlarged. The equipment listed below is provided.

2 Graphlex cameras, 4 × 5.
1 Hand Held camera, 4 × 5, Eastman, A-1 Navy Standard.
2 Eastman 3-A kodaks with anastigmatic lens.
2 Eastman 3½ × 8½ view kodaks.
1 Moving picture kodak, B. and H.
1 Enlarging outfit, 8 × 10 with Cooper Hewitt light.
1 Enlarging outfit, 5 × 7.
1 Contact printer.
1 Dark room.
1 Moving picture machine for exhibiting official pictures.
1 Moving picture developing tank.
1 Moving picture printing machine

PIGEONS.

On the poop deck at the stern there is a pigeon house measuring 10 × 16 × 6 feet, separated into two compartments with a passage between. This house is designed for 48 pigeons. The training of pigeons for use on shipboard presents an interesting problem quite different from the problem of training the birds for shore use. Though it was at first thought possible to train them to return to any ship in the locality from which it was taken, it has later been found feasible always to keep a number of untrained birds for release among new surroundings.

Much experimental work with pigeons however remains to be accomplished. Whether they return to the vessel or to a naval port where the vessel has been stationed, their utility in case of accident will be vindicated by the information that the pigeons may bring to those not connected with this vessel.

AEROLOGICAL FACILITIES.

The aerological laboratory is a detached house on top of the flight officers' quarters and measures 12 × 8 × 8 feet. There are two aerographers assigned to the vessel who issue a daily bulletin of flying weather. The equipment of this laboratory is:

1 Barograph, recording weekly.
1 Hydrograph, recording weekly.
1 Thermograph, recording weekly.
1 Theodolite.
1 Davis hand anemometer.
2 Mast anemometers.
1 Nephoscope.
1 Pilot balloon balance.
1 Pilot balloon filling outfit including:
  Filling valves and Hydrogen tanks.
2 Split second timers.
1 Nemescope.

INTELLIGENCE OFFICE.

This is a house on top of the forward end of the after superstructure on the same level with the sick bay, but separated therefrom by the athwartship smoke stack. It was originally intended for a place to concentrate all information received from various sources, radio telephones, pigeons and plane observations somewhat in the manner of a central station for fire control, but it is also used for an office for the Senior Flight Officer and for a Communication Office.

MACHINE SHOP.

Located on the second deck forward under the joiner shop, this shop contains the following equipment:

	Horsepower
1 Machine drill press	3
1 Power hack saw	½
1 Small grinder, size 8 × ¾	½
1 Bench grinder	
1 Small tool grinder	½
1 Sheet metal cutter, capacity 3/10-inch metal	2
1 Bending slab	
1 Milling machine	5
1 Lathe, 6 inches × 16 inches swing	3
1 Lathe, 5 feet 6 inches × 10 inches swing	3
1 Lathe, 5 feet 6 inches × 16 inches swing	3
1 Bench lathe	¾
1 Shaper, stroke, 16 inches	3
1 Sensitive upright drill table, 16 inches	¾
1 Press, size 3½ inches	
1 Dividing head milling machine	
1 Adjusting tail block milling machine	
1 Adjusting milling machine	
1 Connecting rod adjusting machine	
1 Bench grinder	½

1	Upright sensitive bench drill	½
1	Bench lathe, size 4 × 5	½
1	Tachometer testing machine	3
2	Scales, 1—2,600 pounds; 1—10,000 pounds.	

AIRCRAFT STOWAGE HOLDS.

The four aircraft holds under the well deck extending to the inner bottoms were converted from the former coal spaces. The forward hold extends from the cofferdam abaft the gasoline space through 16 frame spaces and the next one just abaft this one is one frame larger. Between these two forward holds and the after one is the elevator space. The after two holds are of the same size as the forward ones and extend to the break of the poop deck. The forward hold is about thirty-six feet high from the inner bottom to the well deck, the others on account of the depth of ballast are 32 feet high. The width is 64 feet and the length varies from 40 feet to 42 feet. The after hold is designed particularly for wing and propeller stowage.

Each hold is ventilated by two vent ducts on each side supplied by a 3.5 H. P. vent blower located in adjacent tower.

It is estimated that 50 planes without wings could be stowed in these holds with little difficulty.

ARRESTING GEAR.

By far the most interesting as well as the most important feature of the ship is the arresting gear for stopping planes on the flying deck. For the present, however, the details of this are confidential as are the details of foreign airplane carriers.

ENGINEERING.

The reader is referred to the description of the U. S. Fleet Collier *Jupiter* in the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, Vol. XXV, No. 4, of November, 1913, for a complete description of the engineering installation.

ENGINE ROOM.

The engine room is essentially the same in the reference above but there have been certain improvements which may be mentioned. A new low pressure evaporating plant of 12,000 gallons capacity has been installed. A new 6 ton York Co., ice machine has been installed with an ice making capacity of one half a ton a day.

The former auxiliary condensing plant has been removed and there has been substituted for it an Eagle Boat condensing plant, which serves the auxiliary machinery as well as the dynamos. This plant consists of a cylindrical two-pass surface condenser, a turbine driven, centrifugal, circulating pump, a turbine driven, centrifugal condensate pump, two rado jets and a vertical, simplex hot-well pump.

DYNAMO ROOM.

A new dynamo room has been constructed in what was formerly a coal bunker space on the starboard side of the second deck above the engine room. Instead of three 35 K.W. turbo generators which comprised the original ship's installation for light, power and field excitation for the main generator there are now but two 35 K.W. generators which are used for field excitation only. Three 200 K.W. turbo generators have been added to care for the large increase in electrical energy required for use in the new capacity of the ship as a carrier. Two of these are in the dynamo room where the switchboards are also located.

BOILERS.

The fire room contains three double ended Scotch Boilers of the following demensions:

Length, feet and inches, 23—0.75.
Diameter, feet and inches, 16—3.00.
Working pressure, pounds per square inch, 190.

These are a part of the original installation but they have been converted into oil burning. Four Bureau Standard burners are the equipment of each boiler. Air is obtained through the Howden closed ash pit system which is supplied by 4 Terry turbine blowers of 15,000 cubic feet per minute capacity, located in a blower room on the first platform deck just forward of the Engine Room.

OIL FUEL SYSTEM

Ten inner bottom spaces beneath the aircraft stowage holds and elevator space in addition to the two deep tanks forward of the break in the forecastle are utilized for the stowage of fuel oil. They have a combined capacity of 550,000 gallons. Three inch suction and filling lines lead from the manifolds in the fire room to these tanks.

A service tank of 8,000 gallons capacity is located in the pump room which is forward of and communicates with the fire room. Two duplicate systems for the fuel oil service are provided. Each system consists of two pumps and one "U" tube heater with necessary connections to the fuel oil manifold. One set of pumps and the booster pump is located in the pump room forward and the other set on the after bulkhead of the fire room.

STACKS.

There are two stacks, leading from a rectangular shaped athwartship trunk located just above the after superstructure deck between the sick bay and the Intelligence Office. The uptakes from both ends of Boiler No. 1 and the forward end of Boiler No. 2 empty into the starboard side of the athwartship trunk while the uptakes from both ends of No. 3 and the after end of Boiler No. 2 empty into the port side of the athwartship trunk. The port stack rises from this side of the athwartship trunk from a blister foundation on the ship's side at the poop deck level to about eight feet above the flying deck. It

may be raised or lowered on a horizontal athwartship axis till it is horizontal, with the opening either forward or aft. This is done by an electric motor on the main deck.

At the starboard edge of the superstructure deck the athwartship trunk extends downward and ends in an elliptical opening seven feet by eleven feet between the poop deck and the after superstructure deck at the side of the ship.

Dampers control the outlet of gases to either stack and are so interlocked that when one stack is in use the other is closed. The starboard stack has an additional experimental feature in the stack spray which consists of two jets about six feet from the outer edge through which a spray is projected from a branch off the fire main. The purpose of this spray is to cool the gases of combustion and thus to some extent lighten the "bumps" which are caused by the warm gases rising over the flying deck.

WORK SHOP.

The former machine shop remains. An electrical work shop is located on the second deck just forward of the dynamo room and includes facilities for the stowage and charging of storage batteries.

A foundry opposite the electrical work shop on the same deck contains:

1 Oil burning cupola.
1 Bending slab.
1 Anvil.
1 Forge.
1 Store room.
Work benches.

GUNNERY.

GUNS.

There are 4 5-inch 51 caliber guns, one on each side of the forecastle and one on each side of the poop deck. All guns are equipped for individual control only.

BOMBS.

Below the elevator flat on the starboard side stowage is provided for 275 aeroplane bombs of different types. The 5-inch magazines are adjacent to the bomb room for the forward set of guns and on a specially constructed flat in what was formerly the after trimming tank for the after set of guns.

TORPEDO ROOM.

There is a torpedo room for the stowage, charging and overhaul of aero-torpedoes located under the elevator machinery flat on the port side. This contains stowage for 24 torpedoes, a torpedo charging air compressor and a war head compartment adjacent.

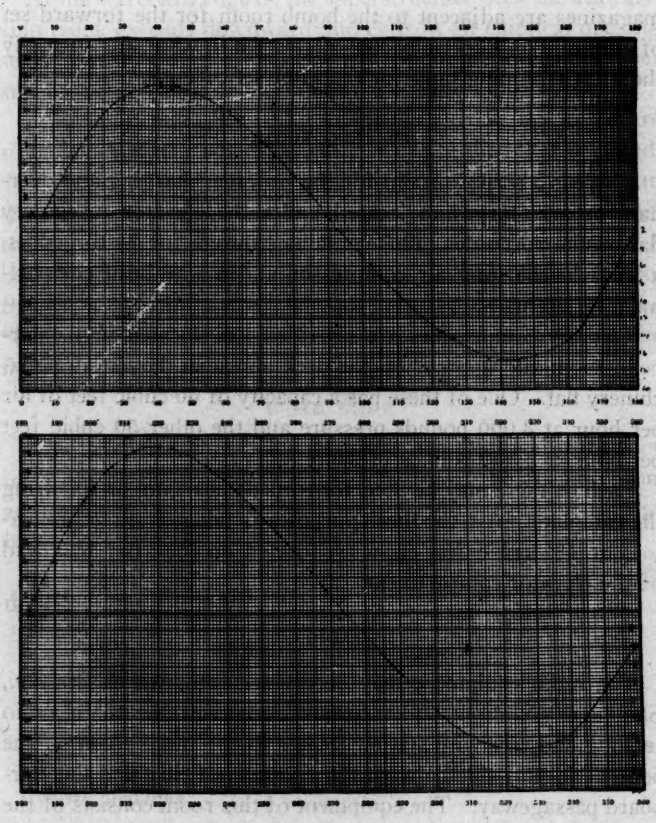
Two air compressors are provided, one located in the torpedo room and another on the deck above in the elevator machinery flat. One of these has a capacity of 50 cubic feet of air per hour at 3,000 pounds pressure and the other 50 cubic feet per hour at 2,500 pounds pressure.

Each of these may be used for torpedo charging or for firing the catapults. Air for these purposes is taken from accumulators of 185 cubic feet capacity, located on the after bulkhead of the elevator compartment.

RADIO.

The radio equipment may be divided into two classes, *i.e.*, Ship and Aviation. The former comprises the main radio sending and receiving station located in the radio room on the poop deck, port side between the Executive Office and the starboard passageway. The equipment of this room consists of the following:

1 Long wave receiver, type S.E. 1899, 1,000—25,000 meters.
1 Seaplane vacuum tube transmitter, 250 W.
1 Medium wave receiver, type 1420 C, 250—7,500 meters.
1 5 K.W. spark transmitter.



RADIO COMPASS CALIBRATION CURVES.

1 Vacuum tube Tel. & Tel. transmitter, type TF 300 W.
1 Motor buzzer transmitter.
1 Portable ½ K.W. spark set for use ashore.

The two motor generators located in the Wardroom passageway for the spark transmitter are operated by distant control from the radio room. The motor generator for the TF transmitter is located near the radio repair room on the second deck.

A radio compass house is located beneath the flying deck and may be raised above that deck by a pulley and weight arrangement. Calibration curves obtained and shown below indicate a high degree of efficiency obtainable with this.

The Aviation Radio equipment consists of:
9 SE 1,300 spark transmitters, 250 W.
1 250 W. portable spark transmitter and receiver.
10 Receiver S.E. 1,414, 200 to 3,500 meters.

A radio workshop and store room is located on the second deck, starboard side, just aft of the break of the poop.

A radio antenna is supported on each side of the ship by two booms, thirty-nine feet long, extending outboard from the well deck on each side at an angle of about 45 degrees. These are for use when the flying deck is rigged for flying and both masts which support the main antenna are housed.

CONCENTRATION SALINITY INDICATORS FOR MEASUREMENT OF SURFACE—CONDENSER LEAKAGE AND DISTILLING PLANT OPERATION.

By Commander C. A. Jones, U. S. N., Member.

Experience has demonstrated the necessity of the elimination of salt from boiler feed water. The importance of this salt elimination increases rapidly with the evaporative capacity per square foot of heating surface.

Salt in the feed water causes priming, scale deposit, and increased corrosion of the boilers and trouble throughout the operating machinery and pipe lines. When the salinity of the water in the boiler is high some salt is carried over with the generated steam. Severe priming may cause serious damage to the turbines and through the formation of water hammers may cause a ruptured steam line. Scale deposit will cause the blistering or burning out of the boiler tubes.

The usual sources of salt in the feed water are the leakage of surface condensers and high salinity of the distillate from the evaporators. The effect of any leakage is cumulative so that a small leak, if continued, results in a considerable increase in the density of the boiler water. It is, therefore, of extreme importance that any leakage be promptly located and checked.

The usual methods of determining the salt concentration are to utilize the salinometer or the silver nitrate chemical method. The disadvantage of these methods is that they do not give a continuous indication; considerable time and care is necessary for an accurate determination, and there is difficulty in getting the operating personnel to obtain tests at sufficiently close intervals to be of real value.

The development and application of the electrolytic concentration indicator has overcome the disadvantages of the salinometer and chemical tests, thereby greatly increasing the ease with which accurate measurements of salinity may be made. Several types of electrical devices for salinity determinations have been developed. The Navy installations have so far been confined to the alternating current, Wheatstone bridge type with the conductivity cells installed directly in the pipe line carrying the water that is to be measured for salinity. The concentration indicator has been developed to meet service conditions by the Babcock and Wilcox Company and the Leeds and Northrup Company in cooperation with the Bureau of Engineering and has now been authorized for all capital ships. The first installation was given a service test on the U. S. S. *Tennessee*. Hundreds of careful, accurate checks were made and it was found that the readings were more accurate than that obtained by the shipboard chemical testing outfit. Improvements since that time have consisted in improving the details of construction. Plans are now underway for the development of a satisfactory system, but one not so expensive or extensive, for destroyers and lighter craft.

The principle of the concentration indicator is based on the variation of resistance offered to the flow of electric current through water of varying salinity. Figure 1 shows graphically the decrease of resistance with increase of salinity. An electrolytic cell consisting of two small metal plates, called electrodes, and immersed in the water to be tested, is connected to an alternating current, Wheatstone bridge for determining the resistance. Alternating current is used to minimize electrolysis of the electrodes and polarization errors.

Figure 2 shows the simplified diagram of connections for the alternating current, Wheatstone bridge. The variable resistance at "T" is to compensate for changes in the temperature of the water that is to be measured for salinity. An increase of 1 degree F. in temperature will cause a decrease of 1.2 per cent in the measured resistance. By adjusting "T", errors due

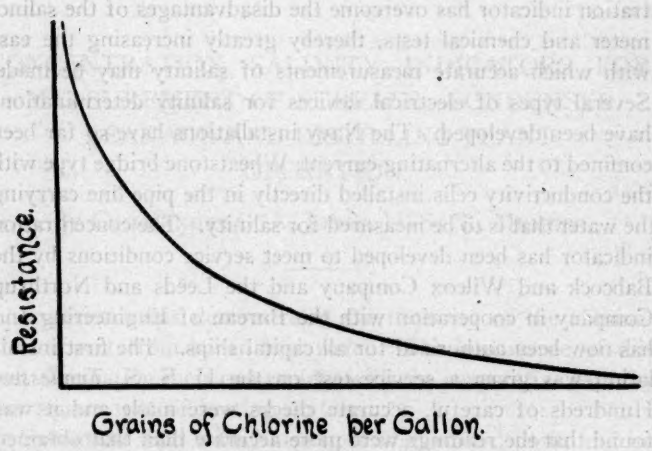

Resistance.
Grains of Chlorine per Gallon.

FIG. 1.

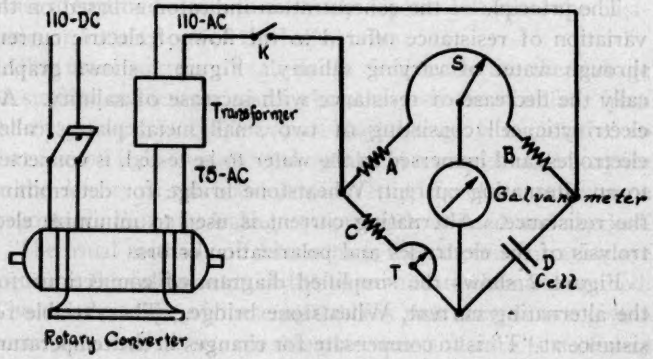

110-DC
110-AC
K
Transformer
T5-AC
Rotary Converter
S
A
B
Galvanometer
C
T
D
Cell

FIG. 2.

to changes of temperature are eliminated. To measure the resistance of the cell the switch "K" is closed and the slider "S" moved until a position is found for which the galvanometer does not deflect from its zero position. The scale of the instrument is expressed directly in grains of chlorine per gallon or in 32's depending on whether the instrument is to be used for condensate or distillate measurements or for brine concentration.

The alternating current is secured from a rotary converter connected to the ship's direct current supply. The alternating current is passed through a transformer before connecting to the Wheatstone bridge. A separate transformer is used for each instrument in order to separate the measuring circuits from the supply, so that no cross currents will result from a grounded ship's circuit or a ground in another concentration indicator supplied from the same rotary converter. Due to the use of the Wheatstone bridge principle, the measurement is independent of line voltage.

The leads to the cells may be of practically any length provided their resistance is small compared to the cell resistance and the resistance placed in the circuit for limiting the current. The leads are run using 4,000 circular mil (No. 14 B. & S. gauge) electric wire. Care must be exercised in making the connections between the cell terminals and the instrument terminals to prevent a high resistance due to poor contact. The standard cell is made so that it can be removed for examination and cleaning without interrupting the flow in the pipe line. Figure 3 shows a section through the electrolytic cell and removal device. For removal the cell is withdrawn past the gate valve, the valve closed, and the ground union broken.

The electrodes of condensate and distillate cells are mounted close together due to the high resistance and low salt content of the water encountered. Cells for measurements of high salinity such as the evaporator brine have the electrodes mounted further apart. It is important that the location of the cell be such that it is always totally immersed in the water to be measured

for salinity. The electrodes of the cell and the porcelain to which the electrodes are fastened are protected by a brass cylindrical case. This case also serves the purpose of controlling the velocity of the water over the electrodes. It has been found

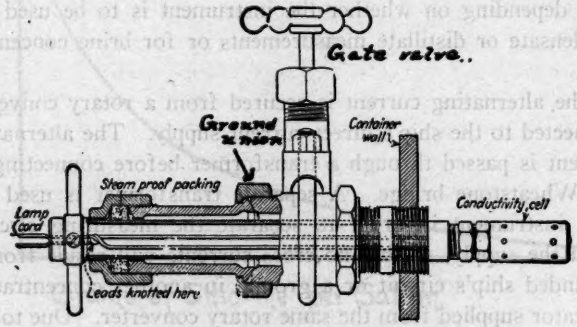


FIG. 3.—SECTION THROUGH CONDUCTIVITY—CELL REMOVAL DEVICE.

from experiment that the velocity of the water over the electrodes has an important bearing on the accuracy of the instrument.

Figure 4 shows the wall type concentration indicator and internal wiring diagram for brine and distillate measurements. Figure 5 shows the wall type concentration indicator and internal wiring diagram for condensate measurements.

The method of operation for taking brine and distillate measurements on the indicator shown in Figure 4 is as follows:

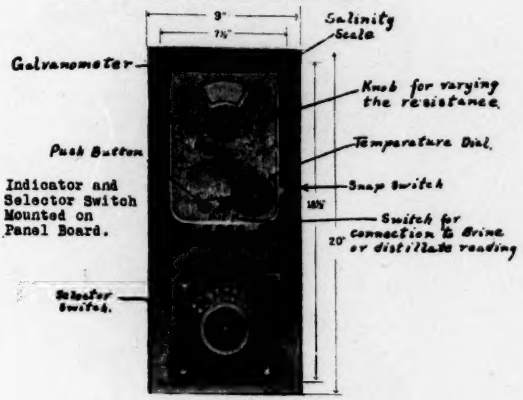
(1) Close the snap switch which supplies power from the rotary converter through the transformer.

(2) Set the selector switch to the position corresponding to the location of the cell in the water to be measured for salinity.

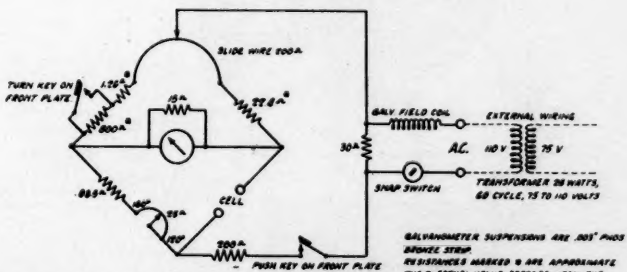
(3) Throw switch so that arrow points to "Brine" if brine concentration is to be measured or to "Dist." if distillate measurements are to be made.

WALL TYPE CONCENTRATION INDICATOR

FOR BRINE AND DISTILLATE MEASUREMENTS



Internal Wiring Diagram of Indicator

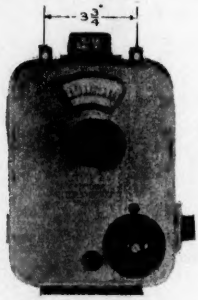


RANGES:-	SALINITY (Grs./Gal.)		RESISTANCE (OHMS) APPROX.	
	Low	High	Low	High
DISTILLATE (SWITCH "A" CLOSED)	0	18	22,000	198
	SALINITY (Grs./Gal.)		RESISTANCE (OHMS) APPROX.	
	Low	High	Low	High.
BRINE (SWITCH "A" OPEN)	0	6	98	2

FIG. 4.

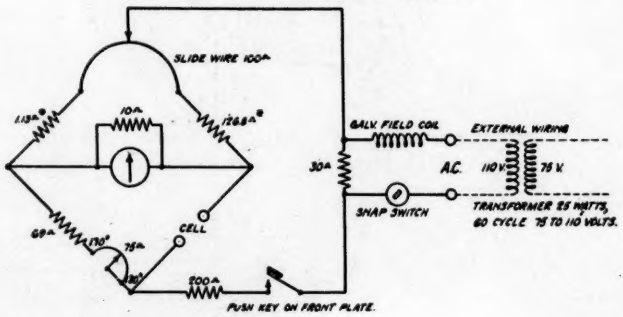
WALL TYPE CONCENTRATION INDICATOR

FOR CONDENSATE MEASUREMENTS



Indicator

Internal Wiring Diagram of Indicator



RANGE: SALINITY ($^{gr}/_{gal}$) = 0 TO 10

RESISTANCE (OHMS) = 126 TO 20000 APPROX.

RESISTANCES MARKED * ARE APPROXIMATE AND DEPEND UPON THE RESISTANCE OF THE SLIDE WIRE

FIG. 5.

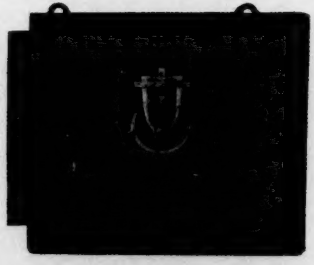


FIG. 6.
SIGNAL LIGHT CONTROLLER.

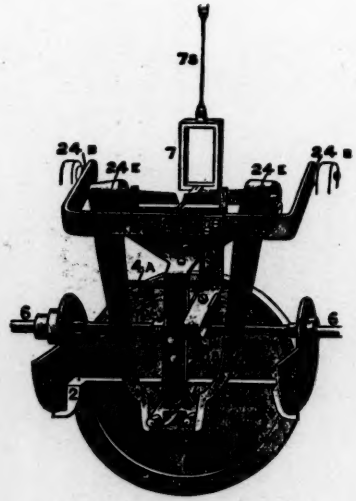


FIG. 8.

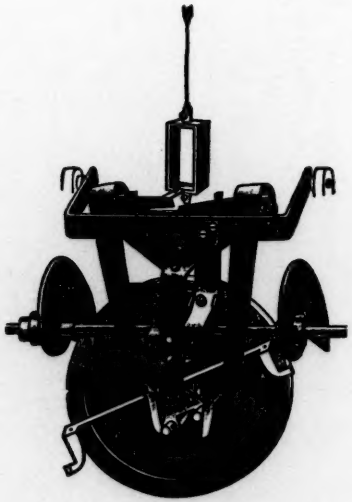


FIG. 9.

(4) Set the temperature compensating dial to correspond to the temperature of the water the salinity of which is to be measured.

(5) Push in push buttom on front of indicator, holding it in, and turn knob for varying resistance until galvanometer pointer comes to rest at its zero position.

(6) Read the salinity scale under its index line. The reading is in "grains of chlorine per gallon" for distillates and in "32's" for brine concentration.

The operation of the instrument shown in Figure 5 is similar to that given above.

Where it is of extreme importance to obtain a continuous indication of the salinity of distillate or condensate water an automatic instrument is used called a signal light controller. This instrument is shown in Figure 6. Signal lights which can be installed at any convenient location, independent of the location of the controller, are used to indicate the quality of the distillate or condensate. For distillate measurements the operation is to have a white light correspond to salinity from 0 to .5 grain of chlorine per gallon, a green light to indicate salinity between .5 and 5 grains, and a red light for salinity over 5 grains. These limits were selected so that the white light would indicate water of sufficient purity for boiler feed, the green light for the ship's tanks (drinking, cooking, washing, etc.), and the red light for water unfit for use. If the operator acts promptly when the incorrect colored light shows he can prevent the salting up of the reserve feed bottoms or ship's tanks due to temporary priming or wrong operation of the evaporators. For condensate, the white light corresponds to a salinity from 0 to .3 grains chlorine per gallon, a green light from .3 to 1 grain, and a red light for a salinity over 1 grain. These limits were selected so that a white light would indicate a perfectly tight condenser, a green light when a small leak was taking place necessitating examination when convenient, and a red light to indicate a bad leak such as a split tube which would necessitate immediate shut down (except in extreme emergency) for re-

pair. This instrument is also equipped with a scale with a sliding pointer to indicate the exact salinity at any instant. Temperature compensation is provided for by a compensating dial mounted beside the signal lights.

The controller consists of three distinct parts mounted together as one unit. First, the measuring circuit—the Wheatstone bridge; second, the motor and governor by which the mechanism is driven at constant speed; third, the automatic mechanism by which the bridge is kept balanced and the contacts in the signal light circuits are operated.

The operation of the measuring circuit is similar to that shown for the wall type instrument except that the deflection of the galvanometer causes the automatic balancing of the Wheatstone bridge.

Figure 7 shows the motor and governor for driving the mechanism at constant speed. The motor is a series motor and the speed is controlled by a flyball governor opening and closing a shunt around a resistance in the motor circuit. The governor weight 45*e* is balanced against the spring 45*g*. For an increase or decrease in the speed, contact at "G" is opened or closed, thus inserting or short circuiting resistance in the motor circuit thereby causing a decrease or increase in speed. The motor speed may be adjusted to its correct value by adjusting the tension on spring 45*g* by screw 45*j*. The correct value of motor speed is 27½ revolutions per minute of shaft 6.

Figures 7, 8 and 9 show the automatic mechanism by which the bridge is kept balanced and the contacts in the signal light circuits are operated. The movements of the galvanometer system 7, which swings about the vertical axis 7*s*, are controlled by the electrical conditions in the measuring circuit. When the galvanometer system is at rest (*i.e.*, the measuring circuit balanced), the galvanometer pointer lies directly under the space between the ends of the two right angle levers 4L and 4R, which are held pivoted at 24E and 24E. From this position of rest, the galvanometer system may deflect under the influence of current in the galvanometer, until its pointer lies at any position between the two stops on the rocker arm 5.

By the cam 6B on the motor driven shaft 6, the rocker arm, which is pivoted at 24B, is periodically raised, and as it is raised, it picks up the galvanometer pointer and lifts it. If, at that instant, the galvanometer is balanced, the pointer is raised

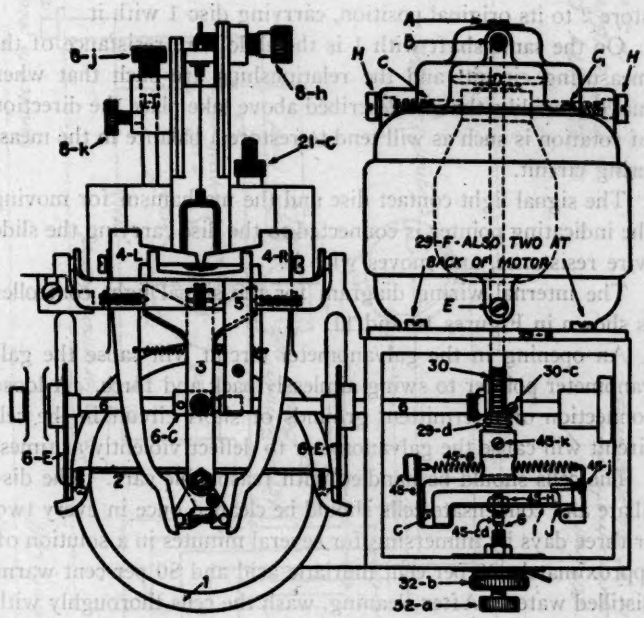


FIG. 7.

into the space between the two levers 4L and 4R. If, however, as the rocker arm is raised the galvanometer is unbalanced and its pointer lies under one or the other of the right angle levers 4L and 4R, which are pivoted at 24E and 24E, the pointer, as it is raised, carries up with it the horizontal side of one of these right angle levers.

The resultant position of the parts of the mechanism is shown in Figure 9. The arm 2, which is tilted from its horizontal position shown in Figure 8 to the position shown in

Figure 9, is one member of a clutch whose other member is the disc 1, and at the instant the movement took place, 1 and 2 were held apart by the cam 6C on the motor driven shaft. As the rocker arm 5 falls, 1 and 2 come into engagement and, the cams 6E and 6E rotating, engage with extension of 2 and restore 2 to its original position, carrying disc 1 with it.

On the same shaft with 1 is the slide wire resistance of the measuring circuit, and the relationships are such that when movements like the one described above take place the direction of rotation is such as will tend to restore a balance in the measuring circuit.

The signal light contact disc and the mechanism for moving the indicating pointer is connected to the disc carrying the slide wire resistance, and moves with it.

The internal wiring diagram for the signal light controller is shown in Figures 10 and 11.

An opening in the galvanometer circuit will cause the galvanometer pointer to swing aimlessly back and forth. A loose connection or intermittent grounds or short circuit in the cell circuit will cause the galvanometer to deflect violently at times.

The cells should be handled with reasonable care. The distillate and condensate cells should be cleaned once in every two or three days by immersing for several minutes in a solution of approximately 20 per cent muriatic acid and 80 per cent warm distilled water. After cleaning, wash the cells thoroughly with water to remove all acid. The brine cells should be cleaned in the same manner, about every four hours.

The cell electrodes are platinized in order to prevent too rapid formation of scale which would cause rapid change in their accuracy. The platinizing also prevents the disintegration of the electrodes. After cleaning with muriatic acid solution their surface should be examined and if properly platinized their surface should show a uniform velvety black appearance. If the bright platinum shows through they should be replatinized. To replatinize the following procedure is followed:

SIGNAL LIGHT CONTROLLER FOR DISTILLATE AND CONDENSATE.

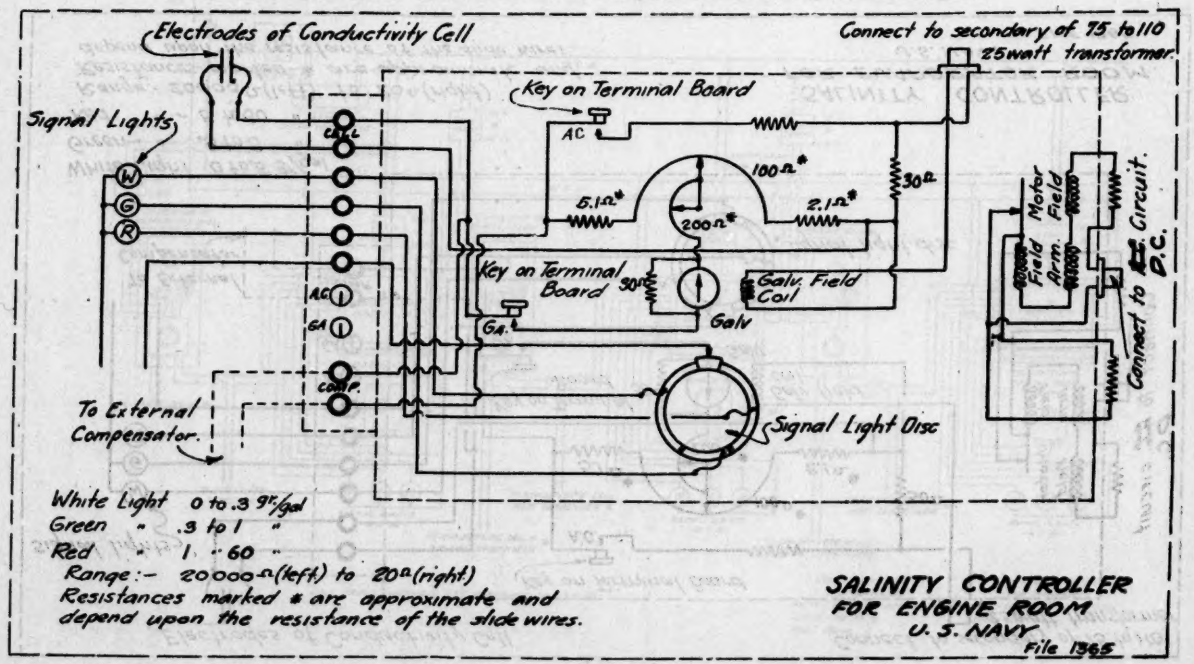


FIG. 10.—INTERNAL WIRING DIAGRAM OF CONTROLLER FOR ENGINE ROOM.

SIGNAL LIGHT CONTROLLER FOR DISTILLATE AND CONDENSATE.

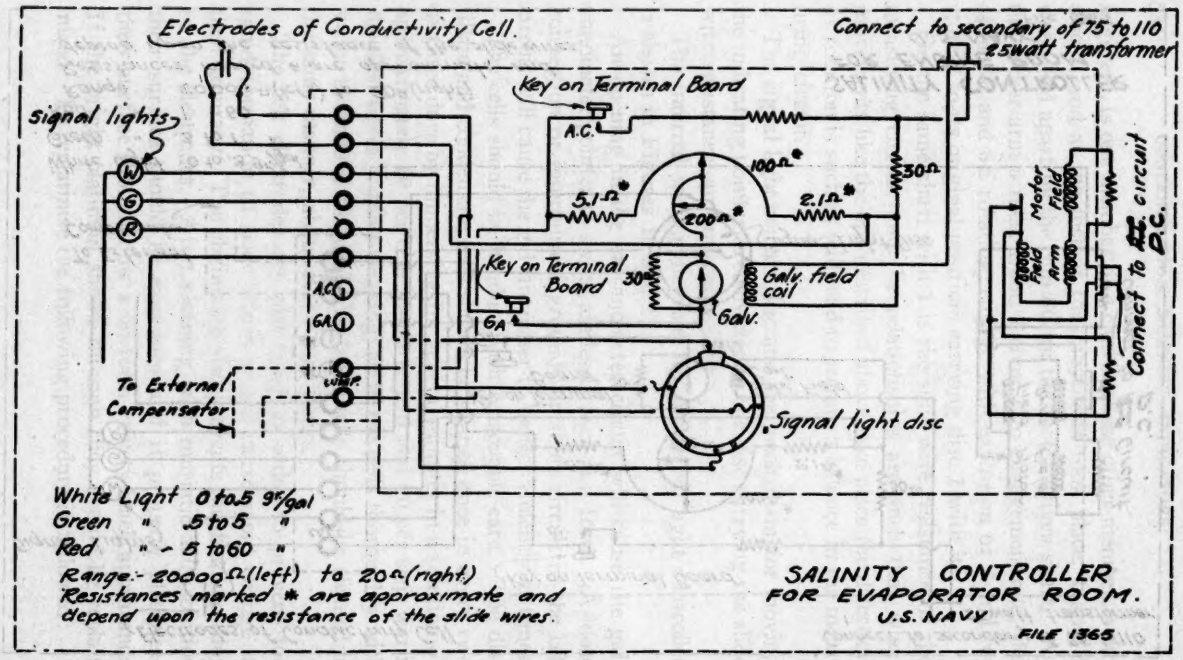


FIG. 11.—INTERNAL WIRING DIAGRAM OF CONTROLLER FOR EVAPORATOR ROOM.

FIG. 11.—INTERNAL WIRING DIAGRAM OF CONTROLLER FOR EVAPORATOR ROOM.

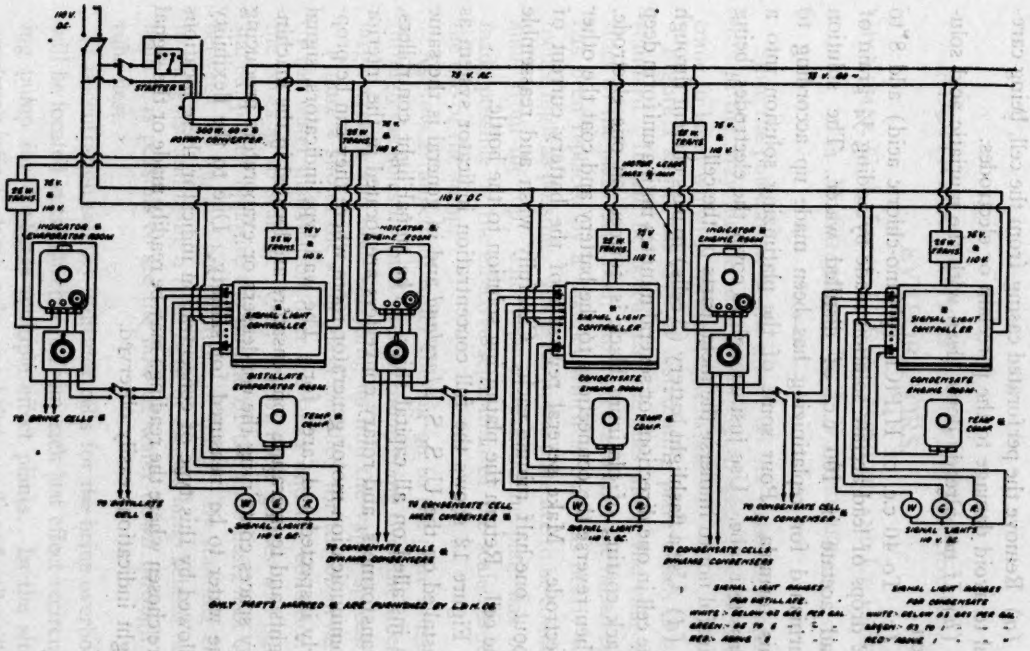


FIG. 12.

(*a*) Remove the perforated casing from the cell, being careful to avoid damage to the porcelain or electrodes.

(*b*) If not already done, clean with the muriatic acid solution.

(*c*) To 40 c.c. of H_2PtCl_6 (platino-chloric acid) add 8 to 12 drops of lead acetate solution made by adding ¼ grain of lead acetate in 100 c.c. of distilled water. The solution furnished for replatinizing has been made up according to this formula. Pour some of the platinizing solution into a glass test tube. Use just enough to cover the electrodes, being careful not to immerse the metal parts of the cell.

(*d*) Use a flashlight battery (3 cells) to pass current through the cell in one direction for several minutes until a uniform deep black coating of platinum black is formed on one electrode. Then reverse the connections to the battery and coat the other electrode. Make several reversals of the battery current of about one-half minute each. Carefully wash and reassemble the cell. Return the platinizing solution to the bottle.

Figure 12 shows the full concentration indicator system as installed on the U. S. S. *Maryland* and in general is the same as installed on all capital ships. The signal light controllers, transformers, and rotary converters are located in the interior communication motor generator room where they can be properly inspected and cared for. The wall type indicators, signal lights and temperature compensators are located in the machinery spaces containing the condensers or evaporators producing the water to be measured for salinity. Due to the flexibility allowed by this type of concentration indicator these locations are chosen where the readings may be readily made or the signal light indications easily observed.

ADDITIONAL NOTES ON LOW PRESSURE EVAPORATORS.

By Comdr. C. A. Jones, U. S. N., Member, and Lt. Comdr. A. M. Charlton, U. S. N., Member.

Since the publication of the article on low pressure evaporators in the May number of the Journal, additional data has come to hand which is incorporated herein.

Figure 1 shows the type of distributing plate which is now standard. It will be noted that the slits along the ends and sides of the plate have been eliminated and that the raised edge has been replaced by a channel which is made as nearly to 2 inches high as possible. All holes in the plate are to be countersunk from the top. This type of distributing plate has been made possible by the effective separation of the scale through the use of the sump.

As now arranged all water coming on to the plate must go through the holes in small streams, none going over the sides in large streams and none splashing up into the vapor spaces.

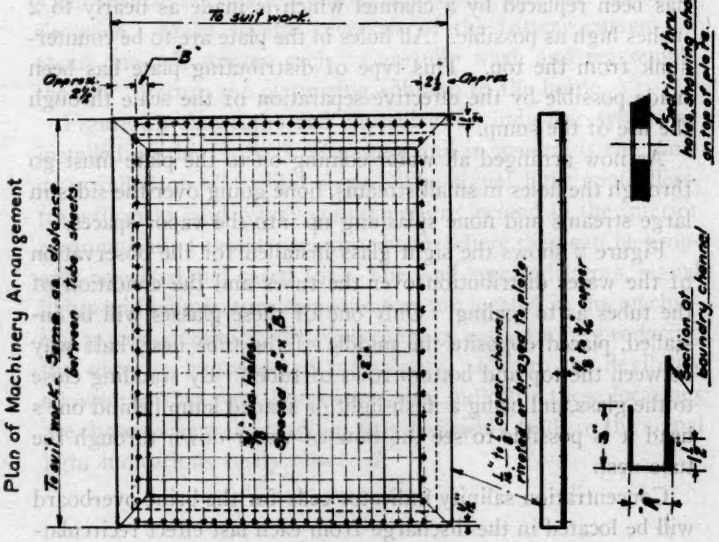
Figure 2 shows the sight glass installed for the observation of the water distribution over the tubes and the condition of the tubes as to scaling. Only one of these glasses will be installed, placed opposite the middle of the tube nest, half way between the top and bottom rows of tubes. By standing close to the glass and using a flash-light or shaded lamp behind one's head it is possible to see the flow of water down through the tube nest.

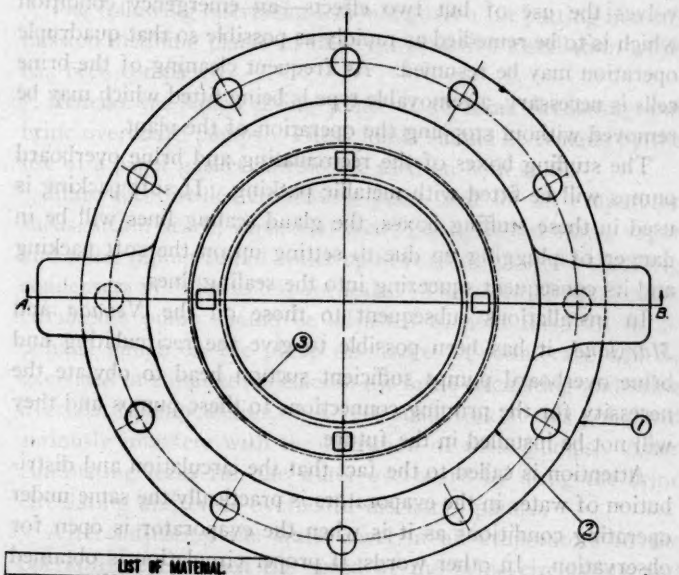
Concentration salinity indicator cells for the brine overboard will be located in the discharge from each last effect recirculating pump for duplicate multiple effect plants. In the single quadruple effect plants only one brine overboard cell will be fitted and that in the discharge of the brine overboard pump. Only one cell is necessary here as double effect operation in-

Note:- Dimension "A" to be as near 2" as possible - limited by clearance for withdrawal of tube nest and maintaining a minimum of 1½" between distributing plate and top of upper row of tubes.

Dimension "B" to be such that a row of holes is over each tube of the upper two rows of tube nest

Fig 1. Distributing Plate. Modified Lillie Type Evaporator.

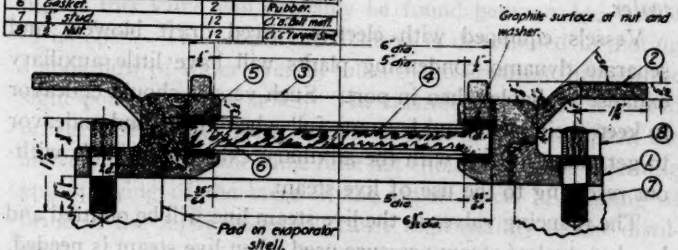




LIST OF MATERIAL.

QUANTITIES ARE PER One Port Light

PIECE NUMBER	NAME OF PIECE.	NUMBER WANTED	SPARE.	MATERIAL.
1	Flange.	1		Comp.'G'
2	Port Light Holder.	1		Comp.'G'
3	Port Light Holder Nut.	1		Comp.'G'
4	Port Light.	1		Glass.
5	Washer.	1		Sheet Brass.
6	Gasket.	2		Rubber.
7	⅞ Stud.	12		Ø.B. Bolt matl.
8	⅞ Nut.	12		Ø.C. Forged Steel



SECTION ON A-B.

Fig. 2. EVAPORATOR PORT LIGHT

volves the use of but two effects—an emergency condition which is to be remedied as rapidly as possible so that quadruple operation may be resumed. As frequent cleaning of the brine cells is necessary, a removable type is being fitted which may be removed without stopping the operation of the plant.

The stuffing boxes of the recirculating and brine overboard pumps will be fitted with metallic packing. If soft packing is used in these stuffing boxes, the gland sealing lines will be in danger of plugging up due to setting up on the soft packing and its consequent squeezing into the sealing lines.

In installations subsequent to those on the *Nevada* and *Maryland,* it has been possible to give the recirculating and brine overboard pumps sufficient suction head to obviate the necessity for the priming connections to these pumps and they will not be installed in the future.

Attention is called to the fact that the circulation and distribution of water in the evaporators is practically the same under operating conditions as it is when the evaporator is open for observation. In other words, if proper circulation is obtained when pumping cold water over the distribution plate with the evaporator open, the same distribution will take place when the water is hot and evaporating, *provided the pump is pumping water.*

Vessels equipped with electric forced draft blowers and separate dynamo condensing plants will have little auxiliary exhaust available when in port. Such vessels should endeavor to keep their tanks and bottoms full when at sea and endeavor to get along in port with the auxiliary exhaust available without resorting to the use of live steam.

The reducing valves in the live steam line will be omitted and hand control of steam pressure used when live steam is needed. The orifice in this line and the relief valve on the coil will protect the tube nest and steam head against full boiler pressure, as the orifice limits the amount of steam coming to the tube nest and the relief valve on the steam head is designed to care for the full capacity of the orifice.

The following instructions should govern for putting in commission distilling plants of this type after the installation work has been completed.

Remake the flange casing joints of all brine circulating and brine overboard pumps. These joints should be remade by the use of a paper gasket and shellac used on both sides.

Start the distiller circulating pump and fill all evaporator shells, steam heads, vapor and water piping steam heads, vapor and water piping full of water up to and including the distilling condensers in order to test for leaks. The speed of the distiller circulating pump should be adjusted to hold from 15 to 20 pounds gauge on the pump discharge. Special care will be exercised in eliminating leaks in the brine circulating and brine overboard pump suctions. A very small leak at this point will seriously interfere with the operation of these pumps. After eliminating leaks run the water overboard by using the brine circulating and brine overboard discharge pumps.

After adjusting the water level in all evaporators to the operating level, check the speed of the brine circulating and brine overboard pumps and set to operate at full normal R.P.M. Now adjust the valve in the brine circulating line for proper distribution of water through the tube nest. The proper opening of this valve will probably be found between 1½ and 3 turns open. The proper distribution is determined by noting a good flow of water over all tubes by observation through the sight ports and man holes. The quantity of water delivered on the distributing plate should be such as to accomplish the above result but at the same time not enough to cause splashing and spraying of the brine in the vapor space. Splashing of brine in the vapor space will cause high salinity of the distillate.

After proper distribution of the brine has been established close up the evaporators and adjust the brine overboard. The amount of brine discharged overboard should be adjusted by the valve on the brine overboard discharge pump. This valve setting under operating conditions should be adjusted to hold

the brine overboard at 2½-32's, ¼ to ½ turn open of this valve will probably effect this.

After the evaporators are closed turn on the steam to the first effect evaporator and then start the distiller air pump operating at slow speed. Until satisfactory operation is established use hand feed regulation (By-passing the automatic feed regulators). All air vent cocks from the evaporators should be opened wide when the air pump is started and kept wide open until the plant is up to its rated capacity. When vapor begins to pass through the distilling condenser adjust the valve in the distilling condenser circulating water overboard discharge line so that the water overboard will be at as high a temperature as possible while still maintaining the proper vacuum. This high temperature of the overboard is advantageous in obtaining high temperature feed.

NOTES.

TEMPERATURES AT WHICH PHYSICAL CHANGES OCCUR.

The temperatures listed below (reproduced from the "Iron Age") are chiefly those relating in some way to the production and heat treatment of steel or to the fusibility of steel and of its components. The higher temperatures—that is, those of vaporization and dissociation of compound substances employed in steel metallurgy—are mostly unknown, and so far as we can see now they would have only a scientific interest for iron and steel men.

In speaking of the melting-point of a substance, either element or compound, one must understand what is meant, as there are a number of temperatures near the melting-point which bear some relation to it. Some of these are practically useful, and some have only a scientific value. These are—

1. The temperature at which the substance sinters, as a clay. This is useful. It is below the fusion point, No. 4.
2. The highest temperature at which a vessel made of the substance can be used. Instance a porcelain crucible. This is useful. It is lower than No. 1.
3. The temperature at which it softens in any degree which, as in a firebrick, limits its usefulness. This is useful. It is lower than the fusion point, No. 4.
4. The temperature at which its edges are rounded. This is often called the fusion point, and is generally the one given in the table. It has only a scientific value.
5. The temperature at which it becomes fluid. Instance a slag. This is interesting, but taken alone is not useful. It is above No. 4.
6. The lowest temperature at which it may be handled molten in practice, as a slag or metal in a metallurgical operation. This is useful. The temperature may be 50 degrees or 100 degrees C. (90 degrees to 180 degrees F.) above No. 5 as the substance leaves the furnace.

Degree Temperature. C.	F.	
−40	−40	Steel containing 25 per cent nickel, which is non-magnetic because of having been heated, is made magnetic; that is, it will be attracted by a magnet.
0	32	Iron and steel have higher tensility and lower ductility than at higher temperatures up to 100 degrees C.
100	212	Water boils. Least tensility of steel below red heat (Howard, French). Greatest contraction of area of boiler steel below red heat (French). When steel containing 0.95 per cent of carbon is hardened and drawn at 100 degrees C. it has the least degree of solubility in 1 per cent sulphuric acid (H_2SO_4) (Heyn and Bauer). When the same steel is dissolved in acid the greatest proportion of the carbon is evolved as

Degree Temperature. C.	F.	
		gaseous hydrocarbons. Sheffield drill rod, having presumably about 1 per cent of carbon, when hardened and drawn at 100 degrees C., has least specific magnetism (Barns and Strouhal). Steel loses its sonority when struck between 100 degrees C. and 150 degrees (Law). The low tensility and high ductility indicate this temperature as being the best for cold-drawing operations on steel.
120	248	Seasoning steel to increase ductility.
150	302	Least ductility of Yorkshire iron rods (7.5 per cent).
204	399	Least ductility of manufactured iron (12 per cent).
220	428	Faintest visible straw color in tempering eutectic steel having 0.9 per cent carbon.
290	554	Greatest tensility of steel and wrought iron.
292	558	Blue color given to eutectic steel in tempering.
350	662	When soft steel (0.07 per cent C.) is quenched and reheated at 350 degrees C. it has the greatest degree of solubility in 1 per cent sulphuric acid. Nickel becomes non-magnetic.
400	752	When hypereutectic steel containing 0.95 per cent carbon is quenched and drawn at 400 degrees C. it has the greatest degree of solubility in 1 per cent sulphuric acid. The steel is by that treatment wholly converted into troostite.
420	788	Zinc melts. Useful for calibrating.
444.5	832	Boiling point of sulphur (for calibrating).
475	887	Lowest visible red of iron in pitch darkness (Howe).
480	896	Greatest strength of cast gun iron (Howard).
580	1076	High nickel steel containing 25 per cent nickel becomes non-magnetic.
655	1211	Aluminum melts.
675	1247	Point Ar. 1 of all steels containing carbon.
680	1256	Mercury boils.
700	1292	Lowest temperature for annealing eutectic steel containing 0.9 per cent carbon.
750	1382	Point Ar. 2 in cooling steel with not more than 0.45 per cent carbon. Iron becomes non-magnetic.
774	1425	Quenching temperature of chromium steel balls having ½ inch diameter.
800	1472	Sodium chloride (common salt) melts.
802	1475	Quenching temperature of chromium steel balls of over ½ inch diameter.
850	1562	Grain made finer in annealing steel containing 0.27 per cent carbon and 0.88 per cent manganese. Phosphorus a maximum. Specific heat (of 0.1657) of iron occurs (Harker).
890	1634	Point Ar. 3 of pure iron.
900	1652	Highest temperature for annealing softest steel containing not over 0.10 per cent carbon. Critical temperature at which softest ingot iron (Armco) is redshot when rolled.
900	1652	Lowest temperature for casehardening (900 to 1000 degrees).
930	1706	Low limit of temperature for forging steel for cannon.
960	1760	Silver melts.
1060	1940	Quenching temperature for manganese steel.
1083	1981	Copper melts. Used as sentinel pyrometers for heating manganese steel for quenching before real pyrometers were available.
1100	2012	Lowest fusion point of cast iron.
1115	2039	Cobalt becomes non-magnetic.

Degree Temperature. C.	F.	
1120	2048	Magnesium boils.
1170	2138	Fusion point of the more fusible acid open-hearth steel slags.
1177	2150	Forging or rolling temperature for high-speed steel.
1200	2192	Burning temperature of clay firebrick.
1204	2200	Temperature for quenching high-speed steel tools (Bethlehem).
1210	2210	For rolling rail steel.
1250	2282	Ferrate of calcium ($CaOFe_2O_3$) melts. Highest melting point of cast iron.
1251	2282	Ferrous silicide (Fe_2Si) melts.
1255	2291	Manganese melts.
1296	2365	Temperature for quenching high-speed steel tools (Arnold).
1325	2417	Fusion point of manganese steel (13 per cent manganese).
1344	2451	Complete fusion of iron containing 0.89 per cent carbon.
1350	2462	Lowest fusion point of simple steels (1350 to 1400 degrees C.).
1360	2480	Sulphate of calcium melts and decomposes.
1378	2512	Fluoride of calcium melts.
1400	2552	Burning temperature for silica bricks.
1419	2586	Ferrous oxide (FeO) melts
1420	2588	Wrought iron welds.
		Burning temperature for magnesia bricks (1400 degrees to 1450 degrees C.).
		Silicon melts.
1443	2629	Ferric silicide (FeSi) melts.
1450	2642	Moulding sands melt (1450 degrees to 1740 degrees).
1452	2646	Nickel melts.
1470	2678	Silicate of manganese ($MnSiO_3$) melts (1470 degrees to 1500 degrees).
1490	2714	Cobalt melts.
1500	2732	Silicon volatilizes (Whitney).
		Silicate of iron ($FeSiO_3$) melts (1500 degrees to 1550 degrees).
1512	2754	Silicate of calcium ($CaSiO_3$) melts.
1520	2768	Casting temperature of crucible tool steel.
1525	2777	Chromium melts.
		Lead boils.
		Highest fusion point of steel.
1530	2786	Iron melts.
1538	2800	Fe_3O_4 melts.
1548	2818	Fe_2O_3 melts.
1550	2822	Lowest operating temperature of open-hearth furnaces and Bessemer converters.
1560	2840	Fireclay bricks melt (1560 degrees to 1725 degrees).
1565	2849	Silicate of magnesium ($MgSiO_3$) melts.
		Bauxite bricks melt (1565 degrees to 1785 degrees).
1650	3000	Highest operating temperature of open-hearth furnaces and Bessemer converters.
1670	3020	Tridymite melts.
1700	3092	Silica bricks melt (1700 degrees to 1750 degrees).
		Titanic acid (TiO_2) melts.
1710	3110	Cristobalite melts.
1740	3164	Koalin melts.
1750	3182	Pure silica melts.
1755	3191	Platinum melts.
1795	3263	Titanium melts.

Degree Temperature. C.	F.	
1820	3308	Bauxite melts.
1900	3452	Manganese boils.
2050	3722	Chromite bricks melt.
		Pure alumina melts.
2059	3738	Chromic oxide (Cr_2O_3) melts.
2165	3929	Magnesia bricks melt.
2180	3956	Chromite melts.
2209	3992	Chromium boils.
		Silicon boils.
		Aluminum boils.
		Graphite made from amorphous carbon (2200 degrees to 2400 degrees).
2220	4028	Carborundum decomposes without melting.
2450	4442	Iron boils.
2500	4532	Zirconium oxide melts.
2560	4622	Molybdenum melts.
2572	4662	Lime (CaO) melts.
2750	4982	Vanadium carbide melts.
2800	5072	Magnesia (MgO) melts.
3176	5750	Tungsten melts.
3600	6512	Molybdenum boils.
3700	6692	Carbon is vaporized.
5200	9392	Tungsten is vaporized.

—"Mechanical World," June 29, 1923.

THE NEW AIRCRAFT CARRIER.

Although they will be kept very secret for a number of reasons, it is tolerably certain that few recent trials will have aroused the interest of those of the new aircraft carrier *Hermes*, which is now practically completed, the first ship to be specially designed for the work. She has a displacement of only about 11,000 tons, with the speed, armament and other characteristics of a light cruiser, while there are many who stoutly maintain that the duties of an aircraft carrier are certain to be such that no ship of under 25,000 tons' displacement could possibly be expected to fulfil them. This being so, it seems a pity that such a long time must elapse between the completion of the *Hermes* and the trials of the reconstructed *Furious*, whose displacement of 23,000 tons, huge flying-on-and-off decks and high speed are regarded as ideal by the big-ship school. The new *Hermes* will be the ninth ship to bear this name in the Royal Navy since 1796, when the first was captured from the Dutch and avenged her late flag by foundering with all hands a few months afterwards. The last *Hermes* was a light cruiser which, after distinguishing herself by having a complete boiler breakdown and trying to sail home under her tarpaulins, became an aeroplane carrier and was torpedoed. On this occasion there would have been many more lives lost had it not been that fear of fire had caused them to stow hundreds of petrol cans on the upper deck; they floated off when she went down and formed really excellent lifebuoys.—"Shipbuilding and Shipping Record," June 7, 1923.

A STREAM-LINE FILTER.

The stream-line filter which Dr. Hele-Shaw, past-president of the Institution of Mechanical Engineers, has invented, promises to have a very wide field of use. A few weeks ago it was described to the Royal Society, and a few days ago it was introduced to the chemical industry. The first idea in the mind of Dr. Hele-Shaw in regard to this apparatus was the treatment of water. As an instance of its effectiveness, it has completely purified, by stream-line filtration, Thames River water and made it suitable for use in ships' boilers again. The writer has seen the apparatus demonstrated, and it is nothing short of remarkable to notice the manner in which oil and water are separated, dyes removed from a solution, leaving only clear water, paraffin oil used for engine-cleaning purposes thoroughly separated from the impurities, and engine crank-case oil with all its impurities brought back again to its original clear state. The liquids to be filtered are passed through packs of special paper—in a 150-gallons-per-hour filter there are 5,000 sheets of this paper—under pressure, the sheets of paper having large and small holes punched in them so that when placed together these holes represent large and small tubes. It would take too long to describe the precise action of the filter here; but its effectiveness has been demonstrated beyond doubt. For the moment, however, attention is mainly being devoted to small filters for laboratory purposes, for Dr. Hele-Shaw recognizes that the commercial development will follow the more quickly after there has been considerable experience for laboratory purposes. It is said that the cost of renewing the papers in the 150-gallons-per-hour filter is 3s. or 4s., and only a matter of 3d. in the case of the small laboratory filters. There is admittedly a vast field for research in securing the fullest possibilities from this filter, but it undeniably introduces a new and most important principle.—"Mechanical World," June 15, 1923.

THE OSCILLOSCOPE.

A Remarkably Simple Device for Seeing What Cannot Be Seen.

If you held your watch in your left hand, in a darkened room, and held in your right hand an electric lamp which flashed exactly once a minute, and if your first flash showed the second hand upon the figure 12, the second flash and all succeeding flashes would show this second hand in the same position. Hence, it would look to you as though the watch had stopped. If, instead of giving a flash every 60 seconds, your lamp gave a flash at 61 seconds' intervals, you would see that the second hand was rotating at its normal speed of once a minute, since every flash would show the hand as having advanced one second in its regular clock-wise direction. On the other hand, if the flashes come at the rate of 59 per minute, it would appear to you as though the second hand were rotating in the reverse direction, or anti-clock-wise.

These principles underlie a very ingenious and useful instrument called the oscilloscope, which forms the subject of the present story. Its object is to enable the observer to watch the movement of a machine or its parts, which are under such rapid motion as to present a mere blur to the unaided vision. Essentially, it consists of a special form of flash lamp and a gear-box which is attached to some moving part of the machine to be investigated and is also connected to the make-and-break gear which regulates the periodicity of the lamp flashes.

A machine at rest, and the same moving at anywhere from 1,000 to 20,000 revolutions or reciprocations per minute, present two very different problems. An engineer can see the details of his machine when it is at

rest, and the various parts will be in exactly the position and will present exactly the forms as designed; but when the machine is running at high speed, certain undesirable distortions and vibrations may occur. He knows that they are there, his trained ear and his sense of touch tell him this; but because of the high speed of the machine he cannot see their nature or extent.

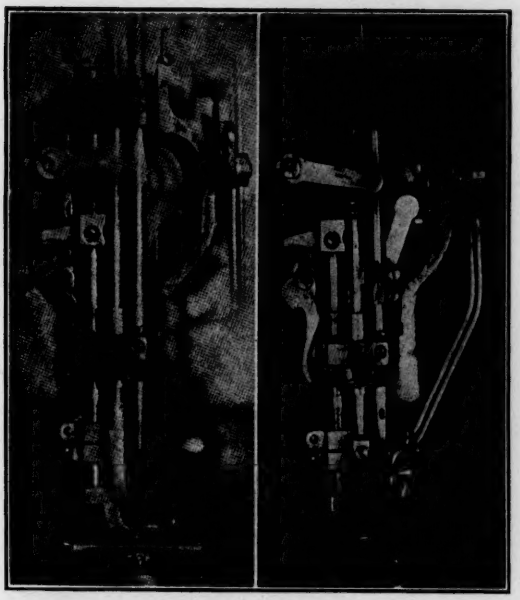
Now the oscilloscope has been designed to enable him to see these various distortions, vibrations and other irregularities. It does this by slowing down the motion; not the motion of the machine, but the apparent motion, as thrown on the retina of his eye. Although the machine may be running at 1,000 revolutions per minute, it is made to appear to him as though it were running at only 10 revolutions per minute, or 1 per cent of its true speed, or even as though it were absolutely at rest.

The means by which this is accomplished are simple and easily understood. First there is a lamp of special construction filled with neon gas, which gives a flash of sharp definition and immediate intensity, without the preliminary red glow and after glow of the ordinary lamp filament. On the gear-box, which is driven by the machine to be examined, is a switch by which the observer can vary the speed ratio between the mechanism to be tested and the flashes of the lamp. This gear-box can control flashes up to 28,000 per minute; so that practically every known variety of high-speed machines can be seen and studied in all its parts, as though it were either moving slowly or were at rest. If the gear attached to the machine has 99 teeth, and it meshes with a wheel connected with the flashing lamp which has 100 teeth, each flash will show the object at an advance of 1/100 of a revolution since the previous flash. Whereas, if the flashes are so controlled that they occur actually at the speed of the shaft, the mechanism will appear stationary in any position desired. A protractor scale provided on the gear-box makes it possible to take actual point-to-point measurements under working conditions.

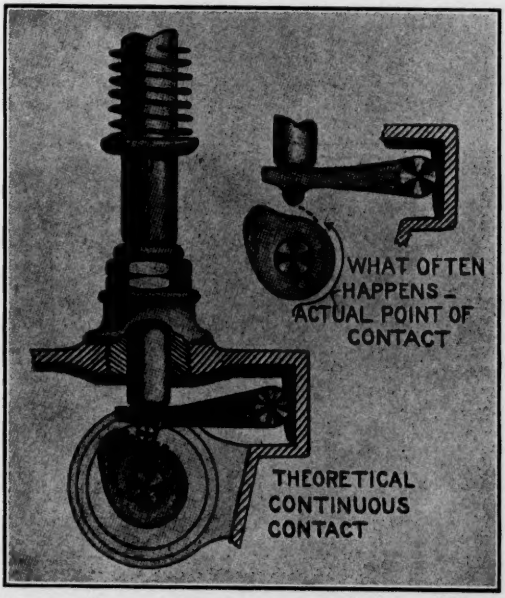
We present herewith two views of the headstock of the same sewing machine. The photograph to the left shows the machine running at a speed of 1,500 revolutions per minute, and the recoprocating arm near the top and the two vertical shafts present the usual shadowy appearance resulting from the high speed. The photograph to the right was taken by the light of the oscilloscope lamp, which was so geared to the machine that all flashes showed the machine at the same exact point in its oscillation; with the result that although it is running at 1,500 revolutions, it appears to be absolutely still. By manipulation of the speed control, the machine may be made to appear to move at any speed desired.

The advantages of being able to follow the motions of a machine which is running too fast for the unaided eye to follow are obvious. Thus it becomes possible to see the extent to which an unbalanced rotating shaft is vibrating about its true axis. The valves of the internal combustion engine are supposed to be alternately raised and seated with great precision; but in watching their operation through the oscilloscope it is found that they may rebound every other stroke and that, consequently, they become fully seated only after bouncing three or four times.

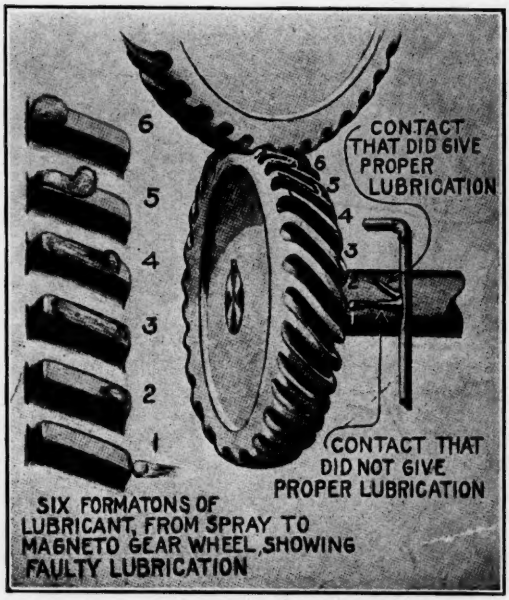
In cotton spinning it becomes possible, by slowing down, to watch the making of the knots and correct inaccuracies. The use of this device in textile printing has proved a great saver of time by making it possible for the operator to look at the moving material without stopping the machine. Another application is in examining the action of lubricants of fast-running machines. Our sketch shows what was discovered to be taking place in the gears of an automobile. The oil delivered to one of the gears was believed to be carried into the other gear, with the result that the whole bearing surface of the two gears was thoroughly lubricated. By slowing down the apparent speed, it was found that the oil was traveling



LEFT: SEWING MACHINE HEADSTOCK, BY DAYLIGHT, AT 1,500 REVOLUTIONS PER MINUTE. RIGHT: SAME HEADSTOCK, AT SAME SPEED, BY RAYS OF OSCILLOSCOPE, WITH MOVING PARTS APPARENTLY STATIONARY.



UPPER RIGHT-HAND SKETCH OF HOW OSCILLOSCOPE SHOWED VALVE STEM, ETC., JUMPING CLEAR OF CAM.



OSCILLOSCOPE SHOWED FAILURE OF OIL TO PASS FROM ONE GEAR TO THE OTHER.

across the width of the cam-shaft gear, the successive stages of its passage being shown in one of our sketches. The effect of this was that the oil was forming in globules on the face of the teeth of the magneto wheel, and none of it was entering the mesh. By changing the position of the delivery pipe, as indicated in the sketch, correct lubrication was at once arrived at.—"Scientific American," June, 1923.

MOTORSHIP *ODENWALD* AND HER NOVEL RUDDER.

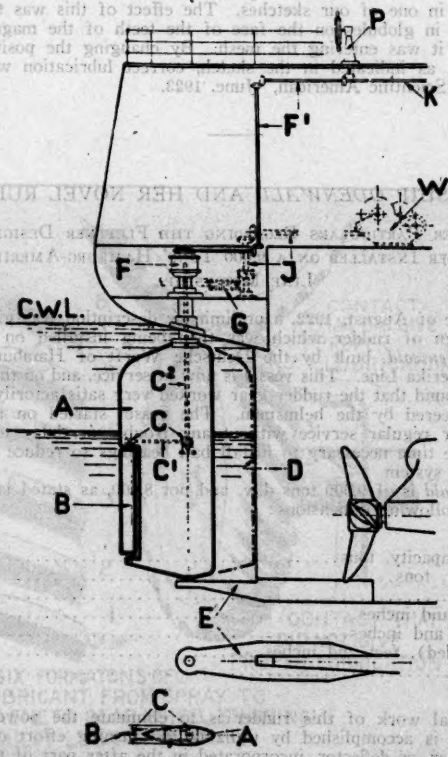
Further Particulars Regarding the Flettner Design of Rudder Installed on a 9,000 Tons' Hamburg-Amerika Line Motorship.

In our issue of August, 1922, a preliminary description was given of the Flettner design of rudder which was then being installed on the cargo motorship *Odenwald*, built by the Deutsche Werft of Hamburg for the Hamburg-Amerika Line. This vessel is now in service, and on the 30 hours' trial it was found that the rudder gear worked very satisfactorily when the vessel was steered by the helmsman. The vessel started on the maiden voyage of her regular service without any trouble or delay to the gear except for the time necessary to install ball bearings to reduce friction in the axiometer system.

The *Odenwald* is of 9,000 tons d.w. and not 8,000, as stated last August, and has the following dimensions:

Deadweight capacity, tons	9,000	
Gross register, tons	5,000	
Power, I.H.P.	3,100	
Length, feet and inches	398	3⅝
Breadth, feet and inches	53	11⅝
Depth (moulded), feet and inches	38	3⅝
Speed, knots	11	

The principal work of this rudder is to eliminate the power steering engine, which is accomplished by utilizing the turning effort of a small auxiliary rudder, or deflector, incorporated in the after part of the rudder, as illustrated by the views given, to move the main rudder. As we already have described this rudder, it is not necessary to go into full details. Briefly, it may be said that the motion of the auxiliary rudder is controlled by means of a mechanical gear consisting of a pair of yokes with horizontal rods running vertically from the hollow axis to the head of the main rudder post. The rudder of the *Odenwald* has about 130 square feet surface, with about 12 square feet surface for the auxiliary rudder. Both are slightly balanced and are fish-shaped in their horizontal sections. It should be noted that the turning gear of the auxiliary rudder is independent of the motion of the main rudder, the latter being able to move completely around, which always occurs when the ship moves astern. Then the rudder automatically turns at 180 degrees and acts as a bow rudder. The steering principle remains as before, acting very promptly under the influence of propeller suction, because the main rudder is already turning under the influence of the astern rotating propellers before motion of the vessel herself has commenced the astern direction.



A—Main rudder.
B—Flettner deflector.
C—Yoke.
C¹—Parallel rods.
C²—Deflector shaft.
D—Position of main rudder astern.
E—Skeg.
F—Gearbox.
G—Spare tiller.
J—Shafting for spare tiller.
K—Shafting for steering wheel on bridge.
P—Flettner steering wheel on the poop.
W—Electric winch as spare steering engine.

THE FLETTNER RUDDER.

In conjunction with the rudder installation another device has been incorporated, namely, the gyroscope compass developed by Dr. Anschutz, by which, as with the Sperry mechanical helmsman, the ship automatically follows the course set by the officer on watch and continually corrected by the gyroscopic electric helmsman. At sea steering is said to be more exact than when in the hands of a human element.—"Motorship," June, 1923.

THE COMPOUND DIESEL ENGINE.

A Discussion of the Limitations of the Compound Internal-Combustion Engine as at Present Developed.

(From an Engineering Correspondent.)

Although in actual practice a small portion of the charge of the Diesel type of engine is burnt at constant volume, the engine itself is really intended to work on a cycle in which the heat addition takes place at constant pressure, and thus, whether in the two- or four-stroke engine, we get an up-stroke during which compression takes place, an oil injection at the top end of this stroke, and a firing stroke during which expansion takes place. In the theoretical investigation of the cycle it is usual to consider the compression as occurring either isothermally or adiabatically and the expansion as adiabatic. If it were possible to exhaust at atmospheric pressure, the heat rejection taking place at constant pressure, the efficiency of the cycle, using adiabatic compression as in the Diesel system, could be written as equal to $1 - \frac{(T_4 - T_1)}{(T_3 - T_2)}$, where T_1, T_2, T_3 and T_4 are respectively the temperatures of air before compression, after combustion and at exhaust, or, in terms of ratio of initial to final pressure (r) equal to $1 - \left(\frac{1}{r}\right)^{\gamma - 1}$. In the actual Diesel engine the heat rejection takes place above atmospheric pressure and at constant volume, and not, as in the above expression, at constant pressure. The cutting-off of the "toe" of the diagram at some point on the expansion line, T_3, alters the expression for the efficiency of this modified pressure volume diagram to $1 - \frac{(T_4 - T_1)}{\gamma(T_3 - T_2)}$. Any "compounding" which can occur must be in the direction of dividing the P.V. diagram horizontally along a line parallel with the base or volume line.

Progress in the United States.

Now the proposals to employ compound in lieu of simple Diesel working, and which have emanated mainly from the United States, seem to imply some line of reasoning analogous to that involved in the compound over the non-condensing steam engine. The products of internal combustion, however, are, of course, non-condensable, and consequently the absolute thermal efficiency of the cycle cannot be extended as it is in the steam engine by condensation, which is a possibility due to the property of the working fluid changing its state, but only by an extension of the compression ratio or ratio of the initial to final pressures. Only a brief description of Mr. Elmer Sperry's work on his compound Diesel has been made public, though it is stated that the United States Government have ordered a trial installation, but it would seem as if the problems involved in the solution of the double-acting engine were simple compared to those of compounding the Diesel while the gain in mechanical simplicity over the latter would be very great. Too much thinking down the lines of the steam engine is very bad policy in internal-combustion engine design as the working fluids are so entirely

different in their properties. It is not at all clear what economical gain is likely to ensue from compounding internal-combustion gas expansion, because from the fundamental expression for efficiency and taking into account the fact that the exhaust must be super-atmospheric in pressure, greater theoretical efficiency in the cycle can only be obtained by going up in initial pressure and not down in the exhaust pressure as it is possible to do in the case of steam. By going up in pressure in the theoretical diagram, the gain, even on the standard air cycle generally used for comparative purposes, becomes very slight after a pressure of about 500 pounds per square inch is reached, this being about the upper limit of pressure usual today, and, in fact, the tendency is to get this point down rather than up. With the actual working gas, owing to the increase in value of the specific heat, the theoretical efficiency is even less than it is on the air cycle. If it is the idea that the high-pressure portion of the expansion be utilized in a small high-pressure cylinder (or cylinders), while a low-pressure cylinder of larger volume be used for the balance, then although the smaller piston area will certainly reduce the total load and the stresses due to pressure, the trouble immediately incurred lies in the fact that the total heat passing into the engine at the higher temperature due to the increased pressure will be confined to a much smaller superficial area, and consequently the stresses due to temperature will be very considerably increased, and these are by far the most difficult to handle. As it is impracticable to arrange for different lengths of stroke, the expansion ratio will be in proportion to the cylinder areas, and with a smaller high-pressure cylinder (or any reasonable number of cylinders) this means that the pressure and temperature of the high-pressure exhaust will be considerably more than if equal cylinder areas were used. But equal cylinder areas mean putting up the high-pressure cylinder pressure stresses, and also subjecting the larger cover to greater temperatures than are at present involved. What the fate of any valve between the two cylinders will be is easy to guess. It must be a fairly large one to handle the gas volume without introducing high gas speeds, and between the one and the other there are seemingly unavoidable troubles.

MR. ELMER SPERRY'S DESIGN.

The Sperry compound Diesel was briefly described in our issue of March 2 last year following a paper by Mr. Elmer Sperry on "Compounding the Combustion Engine," read before the American Society of Mechanical Engineers. This engine is a three-cylinder 4-stroke unit, consisting of two small high-pressure cylinders on one central low-pressure cylinder of considerably greater area. It was suggested that this area might be as much as ten times the area of the high-pressure piston and that an expansion ratio of as much as 120 to 1 might be used, yielding a high efficiency because of the low temperature of the exhaust. Now there is very little object in the I.C. engine in a terminal temperature on the firing stroke prior to release or scavenger-air admission much below 750 degrees F. as the entropy of the gases is then very low. With adiabatic compression and expansion, whether in a simple or a compound Diesel, the available work in the constant pressure cycle *rises* with increase of exhaust temperature for a given number of pressure expansions. If a lower exhaust temperature, say 350 degrees F., be used, the ratio of the position work done during the firing stroke to the negative work done on the compression stroke falls considerably. The advantages of "compounding" may considerably be found to lie in more efficient compression of the cycle air which would thereby reduce the negative work, but it must be remembered that while an I.C. cylinder is inefficient as an air compressor compared with the pure air compressor itself and that at least a two-stage air compressor would be used for compressing to anything over even 60 pounds per square inch, in the air compressor it is desirable to

maintain the air at as low a temperature as possible and to deliver cool air if it can be done, whereas in the Diesel the object is to raise the temperature to a point at which it will ignite the injected oil. One of the greatest advantages of solid injection lies in the fact that it enables combustion to take place at a lower pressure because the cooling effect of the expansion of the blast air for injection necessitates compressing to a higher initial pressure in order to obtain the same firing temperature. Nor is it quite clear what expansion ratio is meant by 120 to 1. Is this a volumetric ratio or a pressure ratio? The pressure expansions in a big marine oil engine are about 15 or 20 to 1. With adiabatic expansion, the expression connecting the initial and final temperatures of expansion, T_3 and T_4, in the formula for efficiency given above is $T_3 = T_4 \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}}$ and with, say, 400, 500, 600 or 700 pounds absolute initial pressure, the theoretical air cycle temperatures at the commencement of expansion were 2,700 degrees, 2,920 degrees, 3,100 degrees and 3,260 degrees F. with an exhaust temperature of 750 degrees F. Varying the exhaust temperature with adiabatic expansion and compression does not affect the absolute thermal efficiency of the cycle, but it does very distinctly modify the proportion of the effective work in the cycle, so that if, say, from 500 pounds' pressure we were to drop the exhaust temperature from 750 degrees to 360 degrees F. we should reduce our net work done, or ratio of $\frac{\text{work done in firing stroke}}{\text{work done in compression stroke}}$ by some 48 per cent because our initial temperature would fall as well, whereas, retaining as before the same ratio of expansion, if we went up in exhaust temperature by, say, 200 degrees, we should increase the net work by some 15 per cent, the initial temperature of expansion rising from 3,100 to 3,460 degrees F. A careful study of the internal combustion engine theoretical P.V. diagram for the constant pressure cycle will reveal the objection to compounding. In the first place, there is not the gain to be obtained, as there was in the steam engine, of avoiding internal cylinder condensation which was one of the chief reasons for subdividing the expansion of the steam. In the second place, increase of pressure expansion involves increase of initial temperature and even with ample cooling a practical limit is reached very shortly after exceeding the maximum pressures now utilized in the simple Diesel engine. Sub-atmospheric working, which would give an increase of pressure expansions with lower exhaust temperatures, and thereby admit of a higher theoretical efficiency in the cycle, is extremely difficult to adapt to the reciprocating I.C. engine, though it seems the obvious method of dealing with the I.C. turbine, but even if the mechanical complication could be overcome the production of the vacuum involves so much additional negative work that the mechanical efficiency of the engine and the ratio of the negative work in the cycle become too great to admit of efficiency.

It cannot be doubted that the mechanical design of the Sperry compound engine is a highly ingenious one—it would not work otherwise—but the difficulties involved are far greater than they are in the case of double-acting engines. The writer, however, prefers to base his scepticism of the advantages of compounding on both the mechanical and the theoretical aspects of the problem. Even assuming that the former can be solved, or even taking a still more generous view that they have been solved, for there is strong evidence that Mr. Sperry has made distinct strides in his design, it is impossible to escape from the fact that, compared with steam, the subdivision of the expansion of the products of internal combustion into two stages instead of one offers no advantages, even if one credits the design with effecting the transfer from one cylinder to the other without loss. Stage compression, of course, does offer an advantage in that as long as the

firing temperature is finally reached the higher the compression which can be obtained without exceeding this temperature the better, but so close are we already to the practical limit of temperature in the Diesel cylinder that there is not very much further scope for development while considerable mechanical complication is certainly involved. Nor again does it pay to adopt the extreme limits of temperature apart from mechanical reasons because of the fact that, owing to the increase in the values of the specific heat of CO_2 particularly, and of the other constituent gases in the working fluid to a lesser extent, the utilization of the upper temperature range is somewhat less efficient than at a lower temperature drop corresponding to the same expansion ratio. This effect does not become so immediately apparent when making comparison on the air cycle basis with a constant value for C_p, but in actual work, its influence tends to reduce the absolute thermal efficiency obtainable from given cycle limits very considerably. A great deal of misunderstanding has often been occasioned by imagining that what was advisable in steam practice could equally well be applied to internal combustion engine practice, but the nature and properties of the working substances are so radically different that the reason for the employment of certain systems in the one may be entirely absent in the case of the other. To no small extent is this the case with the attempt to "compound" I.C. cylinders.—"Shipbuilding and Shipping Record," May 10, 1923.

THE MARTIN-NAVY ALL METAL SCOUT SEAPLANE.

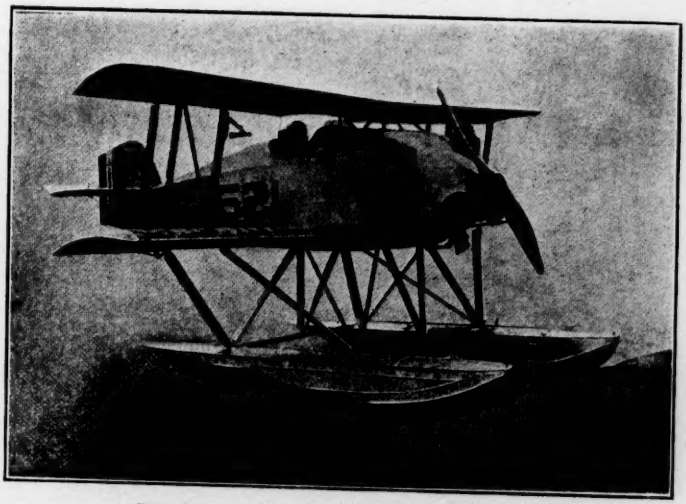
Tests have just been completed on Lake Erie at Cleveland of the Navy's latest seaplane, the MS1, designed by the Navy for shipboard use and developed by the Glenn L. Martin Company. So far as is known, it is the smallest seaplane ever built.

This "mechanical humming bird" is entirely of metal with the exception of the covering, and is only 18 feet wide, 17½ feet long and 7½ feet high from the water line. Its actual weight is less than 650 pounds. Notwithstanding its small size, it is a real airplane, handling and maneuvering in the air as well as much larger planes. While it is not permissible to give actual performances, its speed is quite high, considering that it is a seaplane and that its motor is of low horsepower.

The power plant is a Lawrance, model L4S, three cylinder, air-cooled motor of 60 horsepower. It carries a 6½ foot propeller. The motor is mounted on a vertical bulkhead at the end of a rectangular fuselage built up entirely of steel tubing. The method of assembling the fuselage is worthy of special note. It is built in a jig, the various members being held in place by special clamps until all the fittings have been made. This results in a perfectly rigid structure which requires practically no truing up. The fittings themselves are quite simple being attached in place by rosette welding. This method, which was developed at the Martin factory, consists of drilling holes through both the main members and the fittings and torch-welding the material around the radius of the holes. Tests on this type of fitting have shown exceptional strength while it also allows for a minimum of weight.

The pilot's cockpit is roomy and all the controls and instruments are readily accessible. Outside of radio equipment, no military load is carried.

The tail surfaces and wings are made up entirely of duralumin, channel sections being used largely for the bracing, while the ribs are stamped out of the material in one piece. Two-inch tubular duralumin is used for the wing beams. The leading and trailing edges are of channel duralumin riveted to the ribs. The wings, as well as the fuselage and tail surfaces, are covered with linen.



THE MARTIN-NAVY ALL METAL SCOUT SEAPLANE.

The wing interplane bracing is accomplished by one set of "N" struts on each side. The flying and landing loads are taken by diagonal struts between the floats and the outside of the lower wing. The lower wings are attached to the fuselage bottom longerons, while the upper wings join at a cabane section above the fuselage. This arrangement permits of easy assembling or dismantling in case it is desired to stow the plane in a very small space.

Unique in seaplane construction are the floats. These are entirely of duralumin, the structure being built up of channel section bracing with water tight bulkheads. The float fittings for the brace struts to the plane structure are aluminum alloy castings. Sheet duralumin is used for the float covering. All joints are made water tight by the use of wicking, impregnated with marine glue, placed in the joints at the time of riveting.

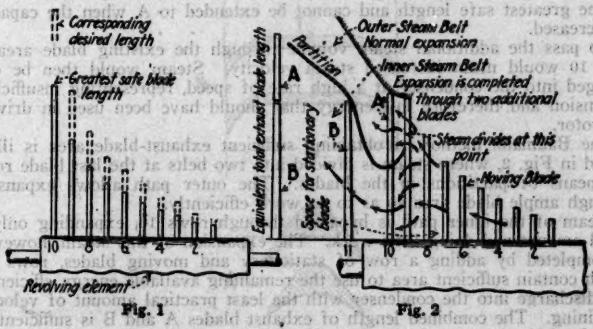
All interplane and float struts are streamlined with sheet duralumin. The gasoline tank is of welded aluminum and holds twelve gallons—sufficient fuel for a flight of two hours at full speed.—"Aerial Age," June, 1923.

RECENT DEVELOPMENTS IN STEAM-TURBINE DESIGN.

Largest Single-Cylinder Single-Flow Reaction Turbine Yet Built.

Description of a turbine built by the Westinghouse Company in which large capacity at moderate peripheral speeds and conservative stresses have been made possible by the use of the Baumann system of blading, while a symmetrically expanding steam path eliminates eddies. Certain constructional features are also described.

The great problem in a large turbine is to provide enough large blades at the exhaust end to pass steam efficiently, while centrifugal force limits



Figs. 1 and 2.—Baumann Principle Increases Capacity of Single-Flow Turbine.

the blade length and diameters to those which will not exceed safe stresses in the materials available.

Designers have hit upon two ways for providing vane area for the complete expansion of the increased volume of the steam and its discharge to the condenser at a moderate velocity. One school has adhered to the single-cylinder direct-flow type, meeting the necessarily high peripheral velocities

by refinements in material, design, and workmanship combined with a reduced factor of safety. Another school has attacked the problem by dividing the exhaust and providing two sets of exhaust blading, one at each end of the cylinder, making what is called the double-flow type; or by dividing the turbine into several turbine-generator units, with a high-speed, high-pressure unit exhausting into one or two lower-speed, low-pressure units. In the latter "cross-compound" type, high speed in the high-pressure unit is made possible by the relatively small high-pressure blades. This means that more work can be obtained efficiently per row of blades, since the work done per blade is approximately proportional to the square of the blade speed, and therefore less blade area is required. The slower low-pressure units are better for the employment of long vanes at the exhaust on account of lower centrifugal force at the required blade speed, and the double-flow principle is easily applied. The cross-compound unit possesses additional advantages of flexibility and reliability, but involves considerably greater first cost. Both of these expedients involve some disadvantages.

For the turbine described here, which is a 35,000-kw. unit under construction by the Westinghouse Co. for the Springdale Station of the West Penn Power Co., a method devised by Karl Baumann, of England, has been adopted.

Suppose that the largest single-flow reaction turbine, as limited by the safe length of the last row of blades, under ordinary design conditions carried 10,000 kw. at the most economical load and it is desired to increase this to 15,000 kw. at practically the same economy.

The high-pressure blades can easily be made longer so as to pass the greater volume of steam at about the same velocity, but the last row of low-pressure vanes is already of greatest safe length and cannot be extended.

Fig. 1 shows schematically such a reaction turbine, the blades in solid lines representing those for the largest single-flow machine of the usual design and dotted extensions suggesting added length for increased capacity, the proportions being only approximate. The last blade, row 10, is already of the greatest safe length and cannot be extended to A when the capacity is increased.

To pass the additional steam volume through the existing blade area in row 10 would mean increased steam velocity. Steam would then be discharged into the condenser at a high rate of speed, representing insufficient expansion and therefore lost energy that should have been used in driving the rotor.

The Baumann method of obtaining sufficient exhaust-blade area is illustrated in Fig. 2, where steam is divided into two belts at the last blade rows by means of partitions in the blades. The outer path allows expansion through ample blade area so as to do work efficiently.

Steam of the inner path is bypassed through rows 10, expanding only a small amount and doing little work. The expansion of this steam, however, is completed by adding a row of stationary and moving blades, rows 11, which contain sufficient area to use the remaining available energy efficiently and discharge into the condenser with the least practical amount of velocity remaining. The combined length of exhaust blades A and B is sufficient to give the desired total exhaust-blade area without an increase in blade length.

Fig. 3 shows in more detail the construction of rows 10 of blades, as well as added rows 11. The outer belt expands normally and efficiently through sections A and B.

The inner belt separated by partitions from the outer, is bypassed through C stationary row 10 with practically no expansion; this section acts merely as a port. Moving section D expands steam to some extent, which helps drive the reaction vanes of the rotor.

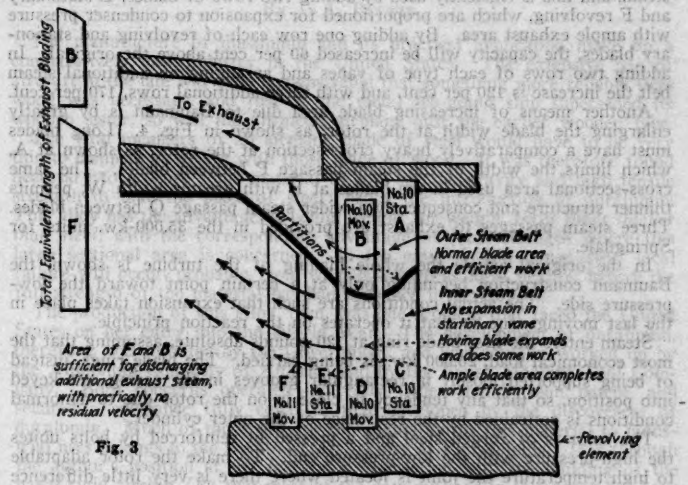

B
F
Total Equivalent Length of Exhaust Blading
To Exhaust
Partitions
No.10 Mov. B
No.10 Sta. A
Outer Steam Belt
Normal blade area and efficient work
Inner Steam Belt
No expansion in stationary vane
Moving blade expands and does some work
Ample blade area completes work efficiently
Area of F and B is sufficient for discharging additional exhaust steam, with practically no residual velocity
F No.11 Mov.
E No.11 Sta.
D No.10 Mov.
C No.10 Sta.
Revolving element

Fig. 3

Fig. 4

FIGS. 3 AND 4.—INNER BELT DOES LESS WORK PER BLADE.

There is still a comparatively great amount of energy remaining in the steam, and this is efficiently used by adding two rows of blades, E stationary and F revolving, which are proportioned for expansion to condenser pressure with ample exhaust area. By adding one row each of revolving and stationary blades, the capacity will be increased 60 per cent above the original. In adding two rows of each type of vanes and arranging an additional steam belt the increase is 120 per cent, and with three additional rows, 170 per cent.

Another means of increasing blade area due to Baumann is by greatly enlarging the blade width at the rotor, as shown in Fig. 4. Long blades must have a comparatively heavy cross-section at the rotor, as shown at A, which limits the width of the steam passage P between blades. The same cross-sectional area used in the blades at B with greater width W, permits thinner structure and consequently a wider steam passage Q between blades. Three steam passages to exhaust are provided in the 35,000-kw. units for Springdale.

In the original article the whole blading of the turbine is shown, the Baumann construction beginning only at a certain point toward the low-pressure side. The actual conditions are such that expansion takes place in the last moving vane so that it operates on the reaction principle.

Steam enters the first blade ring at 120 pounds absolute, assuming that the most economical load, 26,000 kw., is being carried. The blade rings, instead of being supported loosely in rectangular grooves in the casing, are keyed into position, so that any tendency to close in on the rotor due to abnormal conditions is restrained by the resistance of the outer cylinder.

The rotor is of forged steel and a pressed fit reinforced by bolts unites the high-pressure with the low-pressure end. To make the rotor adaptable to high temperature the joint is located where there is very little difference of temperature, and the higher temperature, if any, is effective on the interior member of this pressed fit which would tend to hold it tight in case it became warmer than the recessed member of the fit.

The form of rotating and stationary packing employed is described and illustrated in the original article. In this the rotating element is finished in a number of conical surfaces, while the stationary member contains thin brass annular rings or fins which project very close to the rotating body. Clearances are adjusted by moving the rotating body in an axial direction. If rubbing should occur the conical elements will radiate heat much more rapidly than the old type of ring of cylindrical shape, and there is less liability of stripping than would occur in case thin fins were used in both the moving and stationary elements.

Numerous other features are described in the original article. Of these may be mentioned the emergency governor, which, in accordance with the recent tendencies in design, is of a type that will reset itself automatically while the turbine is running, thus avoiding the excessive slowing down for resetting that was necessary with the older type. In this case the governor includes means for checking approximately whether it operates at the correct speed.

The guaranteed steam consumption of this unit at 300 pounds 650 degrees F. steam at the throttle, with a vacuum of 29 inches of mercury, is as follows:

Load, kw	1,750	26,250	35,000
Lb. per kw.-hr	10.30	9.75	10.15

At the most economical load of 26,000 kw. the following amounts of steam are to be extracted:

High-temperature heater, 12.9 lb. sq. in. abs.: 16,380 lb. per hour.

Low-temperature heater, 3.20 lb. sq. in. abs.: 12,910 lb. per hour. ("Power," vol. 57, No. 20, May 15, 1923, pp. 746–752, 14 figs., *dA.*)—"Mechanical Engineering," July, 1923.

PRACTICAL HEAT TREATMENT OF DURALUMIN.

By J. A. Binnie.

The manufacture of duralumin and its practical application make up a comparatively new industry in the United States. The extensive use of aircraft in the late war soon demonstrated that something more reliable than wood was necessary, and just as the metal sea-going ship superseded its wooden sister, so will the all-metal airplane oust the earlier wooden type.

The vital members of an airplane made of steel would have to be very thin to compete in weight with those made of wood. It was therefore found necessary to procure some material of low specific gravity and high tensile strength to take the place of steel; hence the introduction of duralumin. This alloy has the advantage of being, weight for weight, *three times thicker than steel*, with the corresponding advantage of local strength where this large sectional area is concerned. In the case of thin gauge steel local failure will occur, probably from what is known as local flexure, before the full strength of the material has had a chance to develop.

The production of this alloy, then, was one problem that faced the United States on her entry into the World War. The development of duralumin, therefore, was initiated by the Government, and was actually worked up by the Aluminum Company of America and by the Baush Machine Tool Company. Both concerns operated indirectly under the German patent, and secured a substance fully equal to that developed by the discoverers of duralumin. Much of the credit for the adaptation of this material to industrial uses belongs to the naval aircraft factory at Philadelphia. It is through the author's position at the factory in connection with the building of the naval rigid airship ZR1 that most of the data outlined in this paper has been obtained.

The construction of the airship is the most extensive use of this interesting alloy in the United States. The Navy Department is conducting the construction work through the Bureau of Aeronautics, the Naval Aircraft Factory, and the Naval Air Station at Lakehurst, N. J., where the final assembly and organizing manager are located. The value of such work cannot fail to be reflected in the future commercial employment of duralumin, and marks a new era in the life of American aircraft in general. The all-metal airplane and airship are with us to stay, and the fact that industrial enterprise other than aircraft is deriving the benefit and making full use of duralumin makes a knowledge of this unique alloy of peculiar interest to those engaged in engineering.

Duralumin can now be supplied in all types and sections ranging from rolled plate, sheet, angles, and tubes, to screws and rivets and all forms of castings and forgings. Armor-plate 0.276 inch thick resists penetrations by vertical rifle fire at a minimum range of 1,000 feet. The material is usually furnished in the hard condition. Where large quantities are in stock, it is advisable as a matter of shop routine to paint the annealed stock with some distinguishing color.

How Duralumin May Be Heat-treated.—We will now detail the practical shop method of heat treatment employed on the manufactured stock as it is supplied to the consumer. To make the position clear, the general principles of heat treatment can be enumerated as tempering, annealing, and ageing.

Duralumin is annealed between the temperatures of 350 degrees and 380 degrees C. and cooled in air. On annealing within this range of temperature the metal becomes plastic so that it can be readily worked and formed into all sorts of sections. In the case of complicated or "freak" sections it may be necessary to anneal the metal several times while working. After this process the parts are tempered. This operation consists of immersing the

metal parts in a salt bath heated to a temperature of 480 degrees to 500 degrees C. and then quenching in boiling water. The salts employed are potassium nitrate and sodium nitrate in equal parts. The temperature range for annealing is 350 degrees to 380 degrees C., and for tempering 480 degrees to 500 degrees C., as Fig. 1 illustrates.

The purpose of annealing is to increase the softness and ductility of the metal so that it may be more easily formed into various shapes, such as stampings, and also that it may successfully withstand the strains to which it is to be subjected. The object of the quenching bath is to remove the heat from the metal at a rate best suited for the qualities desired. The bath containing ordinary water (as in this case) has no virtues other than to remove the heat. Recent investigation has demonstrated that air-cooling is slightly better for annealing. The point, however, is not emphasized, as mechanical working tends to harden duralumin as it does other metals.

The tempering process as applied to duralumin might be taken as similar to the process used for hardening steel, with this difference—that steel be-

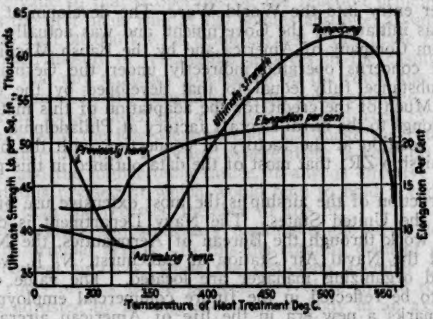


FIG. 1.—HEAT TREATMENT OF DURALUMIN.

comes hard immediately, whereas duralumin becomes soft at first, and then gradually hardens, reaching its greatest hardness in about 100 hours.

Dr. P. D. Merica and his associates have pointed out an apparent analogy between the hardening of duralumin and that of steel: "Most recent thought regards the hardening of steel as due more directly to the suppression of the cementite precipitation (as pearlite), the transformation of γ into α iron having taken place at least in part. The partial suppression therefore of the precipitation of a compound from a solid solution is common both to rapidly cooled steel and to duralumin." This statement of theirs, however, is likely to be misunderstood. It should be borne in mind that the hardness of duralumin increases during the ageing process, which is the precipitation of the compound from a solid solution.

To make this clear let us consider that to draw the temper from steel the metal required heating; if left at room temperature it remains unchanged. The temperature at which duralumin remains unchanged after quenching is that of liquid air (180 degrees C.). Thus ordinary room temperature is a "high" temperature for duralumin—so high that certain internal changes can take place. If duralumin were produced on the moon, where the surroundings are colder than liquid air, it would be necessary to heat duralumin to produce any change in properties.

Effect of Temperature on Strength.—Duralumin heated to 370 degrees C. and quenched remains soft and workable for approximately 24 hours, or if heated to 510 degrees C. and quenched, it also remains soft and workable for the same length of time. It will readily be seen from this that the immediate effect is apparently the same in both instances, but that the ulterior effect on property changes is different. After heat treatment at 510 degrees C. the hardness and ductility increase as actual tests shown in Table I demonstrate. The ultimate strength is in direct relation to the temperature range. It is apparent therefore that the best properties are obtained at 510 degrees C.

TABLE I.

RESULTS OF ANNEALING AND TEMPERING DURALUMIN.

—	Heat ° C.	Quench in Water, ° C.	Ultimate Tensile Strength, Lb. per Sq. In.	Elongation, %	Sclerescope Hardness.
Annealing ..	370	20	35,000	17	18
Tempering .	510	20	55,000	34	20

Dr. Merica's remarks on this subject are enlightening: "It has been shown that when duralumin is rapidly cooled by quenching from temperatures between 250 degrees and 520 degrees C., and aged thereupon at temperatures from 0 degrees to 200 degrees C., the hardness and, at least at lower ageing temperatures, the ductility increase. The actual values of hardness and ductility thus obtained depend upon the quenching temperatures; they increase with that temperature up to about 520 degrees C., corresponding to the increase of $CuAl_2$ in solid solution; at this temperature any free $CuAl_2$ melts as a eutectic and the material is spoiled; this eutectic temperature, therefore, marks the upper limit of the useful quenching temperature range."

From the foregoing remarks it will be evident that from actual scientific tests 520 degrees C. must not be exceeded, otherwise the metal would "burn." In the writer's experience 480 degrees C. has given every satisfaction.

Quenching is usually done in cold water. It has been claimed that quenching is best and most conveniently carried out in boiling water, 100 degrees C. (212 degrees F.), and that the period of ageing is greatly accelerated by the use of boiling water for quenching. In substantiation of this claim it is pointed out that the mechanical properties are better after quenching in hot rather than in cold water, and there is less danger of warping due to cooling stresses.

In the case of tempering this acceleration of ageing appears desirable, but in the case of annealing the benefit seems somewhat doubtful. For instance, the object of annealing is to make the metal treated soft and workable. The longer the metal remains soft, the more operations can be performed without frequent annealing. The rapid ageing of the material therefore would militate against the very properties desired. Accepting this theory, then, it would appear desirable that the annealing process should consist of cooling in air or even in the furnace, for the working life of the metal varies as the ageing period.

Time Period for Heat Treatment.—An important point is the period of time necessary for the material to remain in the bath. Plates to be annealed from 1 mm. to 2 mm. (0.04 inch to 0.08 inch) thick should be left in the

bath for from 10 to 20 minutes at 350 degrees C. For tempering the time is from 7 to 16 minutes at 500 degrees C. The following rough formula for the duration period for heat treatment may prove useful:

$$\text{Time in minutes} = C\sqrt{t};$$

where t is the thickness of the plate in inches, and C a constant (equal to 80 for an annealing heat, and 60 for a quenching heat). This constant C is slightly excessive, and can be taken as a maximum. As an example, take two plates 0.016 inch and 0.070 inch in thickness:

t	$\sqrt{t}$	Time for Quenching in Min. $60\sqrt{t}$	Time for Annealing in Min. $80\sqrt{t}$
0·016	0·1265	7·59	10·12
0·080	0·2828	16·96	22·62

Rivets can have only one form of heat treatment—namely, quenching. About 500 degrees C. is the usual quenching temperature adopted. Great trouble is often experienced because the rivets harden too quickly and subsequently crack. Rivets harden very quickly, due to their small volume and the fact that they must be worked while ageing. They should never be left unworked for more than a few minutes after quenching. It is, in fact, advisable to use them immediately after quenching. If rivets are left unworked for longer than 30 minutes upon riveting, small incipient cracks develop around the periphery of the head. The time allowed in a salt bath can be taken as 20 minutes for a maximum for rivets from 0.09 inch to 0.159 inch in diameter.

The material of which the quenching tank may be composed is of very little importance. A wooden rectangular tank with a sheet-metal lining can be used. It is desirable to have a metal rim, however, if the tank is not lined. A slightly raised perforated platform surrounding the bath is also an asset. The distance from floor to rim need not exceed 2 feet. A convenient size would be 4 feet × 3 feet and 3 feet deep. This size would be suitable for all ordinary small or medium parts. A second bath would then be necessary for the very longest metal parts. Its size would, of course, be governed by the longest metal part to be immersed.

The usual bath accessories would be a sump, inlet and overflow. As rapid filling or emptying is not necessarily called for, these fittings can be of standard type. Where hot water is desired, the bath contents can best be heated with a circulating steampipe coil placed in the interior of the bath, preferably near the base, with a movable perforated tray placed just clear of the coil.

Furnace for Heat Treatment.—A heating chamber usually adopted for the treatment of duralumin is in reality a bath. It can be constructed of cast-iron or of mild-steel plates welded or riveted. Built around the bath is a brick furnace, the requisites of which are as follows:

1. The temperature shall be uniform in all parts.
2. The metal parts shall be heated uniformly on all sides.
3. The temperature shall be readily controlled by the operator.
4. The temperature shall be indicated by a pyrometer.

The methods of heating are electricity, gas, and oil. Charcoal or coke should never be used, because the gases given off, if they should combine

with the nitrates used in the bath, would result in a highly combustible and explosive compound. If gas or oil is used for heating, care should be taken to keep the flame away from the metal bath by means of baffleplates or a brick arch.

A bath of this nature gives ready access to the metal parts being heated, allowing them to be removed quickly and easily. Means must be provided for the support of parts when removing them from the bath after treatment, so that they will not bend or distort under their own weight, nor be damaged by careless handling. Wire-mesh baskets are best suited for this purpose, and the dipping basket should be provided with a long handle. The metal parts to be heated can be nested in the basket. After treatment, the basket with its contents intact is removed from the bath and immediately quenched.

Care should be taken to see that the basket is dry before re-immersing in the molten nitrates. Dripping water from the basket, if falling on the nitrates, will cause innumerable small particles of the molten compound to shoot out of the bath, which would be dangerous to the operator. Blacksmith's tongs or iron hooks can be used to remove the basket from the bath. They, too, should be free from moisture when coming in contact with the molten salts.

It is desirable that the bath should be enclosed on three sides with a hood, leaving the fourth side open for easy access. This construction can best be done with sheet metal, built in the form of a cowl with a ventilator. The fourth side could easily be provided with a hinged door. The object of the extra bath mentioned for small and medium parts is in the nature of an auxiliary, in case one should go out of action, as they frequently do after prolonged use. Another object to be considered is that with only one bath heated to, say, 500 degrees C., a considerable time must elapse before it can be cooled to, say, 300 degrees C., for annealing, or *vice versâ*.

There should be two baths for heating rivets, and these baths will be in use most of the time. The size is small and need not exceed 12 inches × 6 inches × 6 inches. This small size allows them to be quickly heated, which is a decided advantage when it is considered that the bath should be at the desired temperature before the arrival of the workmen. The fifth bath, the one for very large parts, would seldom be in use. It would necessarily be located near the quenching bath of corresponding size. One bath of this size would be sufficient. A shop program could easily be arranged whereby metal parts to be annealed would not jibe with those to be tempered. Sufficient time would thereby be allowed to raise or lower the temperature over a range between 300 degrees and 510 degrees C.

Gauging the Temperature.—Each heat-treatment bath should be supplied with a pyrometer, the indicating dial being placed within the ready view of the operator. It is advisable to procure one or two spare instruments for emergency. It is worthy of note that all heat-treating appliances should be as near the shop as possible, and, for preference, within the shop itself.

Two points to be carefully considered are the utility and the fire risk. The objects of enclosing the bath on all sides are—

1. The heat is concentrated.
2. The range of molten salts is minimized in case of explosion of fires.
3. The salts are prevented from splashing on anyone in close proximity.
4. When it is necessary to reduce the temperature rapidly, an air extractor could be fitted on the end of a flue leading from the cowl.

In the case where it is necessary to heat the bath quickly, and where the bath is not enclosed, a sheet-metal cover can be provided, lined with asbestos on the inner side. The enclosed bath, however, is undoubtedly the best, excluding as it does the entrance of dirt or any foreign, injurious, or dangerous matter.

The number of heating baths is an important item. There should be two small baths for rivets, two for medium parts, and one for very large parts, making five in all. It is possible to have the furnace within the shop, built on a concrete foundation with a surrounding metal guard, and adequately ventilated. There is no denying the fact, however, that where oil burners are used, the fire risk is always present.

The metal can be worked in the cold state, provided there are no very sharp bends and the operation is straightforward. Where members are likely to be subjected to severe working, it is advisable to anneal them. As a matter of interest, the material does not show very good results in fatigue tests, and consequently where the parts are subjected to a great vibration it is advisable to employ steel as a reinforcement.

Effort of Salt Spray on Duralumin.—If duralumin has been properly treated in the pre-ingot stage and is reasonably free from impurities, little or no corrosion is visible if immersed in sea-water. If exposed to salt spray (sea-air) it becomes very quickly coated with a thin film of Al, Fe, and Mg oxides, which are by no means detrimental to the material. On the contrary, they act as a protective coating. When steel has been used in contact with duralumin, very little electrolytic action has been noticeable. It is recommended that copper and brass, which are also strongly electro-negative to duralumin, should never be placed in contact with duralumin where subject to moisture. The metal can be turned and machined at the same speed as brass, and does not seize or drag the tools as do most aluminium alloys.

Kerosene forms a very good tool lubricant in threading or finishing machine parts, leaving a clean-cut surface. When buffed, duralumin takes on a very fine polish similar to silver, except that the surface lasts longer, as it does not tarnish in the presence of sulphuretted hydrogen and similar gases. Riveted members should never be heat-treated on account of warping. Each separate part should be treated before riveting.

Data for Use in Design.—In view of the fact that the full tensile strength of 55,000 pounds can be relied on with confidence, duralumin may be used for any part of a structure where the combination of strength with extreme lightness is desirable. It is worthy of note when working with thin gauge material that a small reduction occurs in the strength surrounding the holes, probably due to the heat produced by drilling. An interesting comparison between duralumin and other metals in the weakening effect of drilling holes for rivets is given in Table II. In the case of duralumin a loss in ultimate strength of about 20 per cent is shown.

TABLE II.

TENSILE TESTS OF PLATE SPECIMENS.

Metal.	Size of Plate.	Area.	Lb.	Lb. per Sq. In.	Elong. in 2in. %	Remarks.
Duralumin	0·65 × 0·100	0·065	3338	51,354	5	0·10in. hole
Duralumin	0·75 × 0·100	0·075	4368	58,240	30	No hole
Aluminium	0·65 × 0·060	0·039	899	21,256	3·6	0·10in. hole
Aluminium	0·75 × 0·060	0·045	896	19,911	6·5	No hole
Steel	0·65 × 0·120	0·078	5376	68,923	13	0·10in. hole
Steel	0·75 × 0·120	0·090	5824	64,711	30	No hole
Brass	0·65 × 0·093	0·0604	3315	54,884	17	0·10in. hole
Brass	0·75 × 0·093	0·0697	3606	51,736	56	No hole
Copper	0·65 × 0·137	0·0890	2689	30,202	11	0·10in. hole
Copper	0·75 × 0·137	0·1027	3468	33,154	45	No hole

Bearing pressure for rivets should not exceed 70,000 pounds per square inch. An ultimate shear stress (in single shear) of 24,000 pounds per square inch may be allowed with confidence. The same relative figure for the double shear can be used as in the case of riveted joints in steel. Where thin-plate members are employed it is preferable to use a large number of small rivets rather than a few of large diameter. In the design of complicated sections care should be exercised to have them uniform—that is, with the free edge falling on the neutral axis. Stresses will then naturally be at a minimum (on the free edge).

A typical stress-strain diagram, the result of research at the naval aircraft factory, is shown in Fig. 2. Note the "humps" on the curve, referred

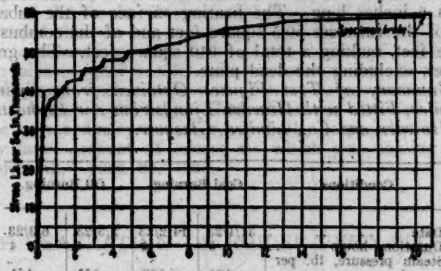


FIG. 2.—HEAT TREATMENT OF DURALUMIN.

to as "false yield-points." This is an exceedingly interesting feature of duralumin under load. The curve is for one specimen tested on a continuous run using a Ewing extensometer up to 8 per cent elongation and a height gauge up to breaking point. When the metal first came into use, a tensile test showed slight extensions at about 20,000 pounds per square inch, giving rise to the belief that the metal had an abnormally low yield-point. Subsequent tests, however, have demonstrated that the metal has several of what have been termed "false yield-points," for want of a better name. They do not continue, as the metal retains its elasticity until the true elastic limit and yield-point occur, usually in the vicinity of 32,000 pounds per square inch. The reason for this false yield feature is somewhat obscure, but has been described as in the nature of molecular adjustments in the material under the action of the load.

From a practical standpoint it can be looked upon as quite a useful property, especially in compression, where struts show much more uniform results, due to the fact that the material adjusts itself to some extent to the load. In the case of steel struts tested variations in the crippling load are always in relation to the homogeneity of material and the uniformity of section.

In conclusion, the author sincerely trusts that the foregoing considerations, while touching only on the more practical features of duralumin, may at least whet the appetite of the nearest engineer for a deeper knowledge of this alloy, which is indispensable to the aeronautical world, and is gradually finding its place in other engineering enterprises.—"Mechanical World," June 22, 1923.

TESTS OF HOWDEN-LJUNGSTRÖM AIR PREHEATER.

The association of the firm of Messrs. James Howden and Co., Limited, of Scotland Street, Glasgow, with the use of heated air and forced draught for marine boilers is of very long standing, and the Howden system has been applied to thousands of vessels containing an aggregate of over 24,000,000 horsepower. The firm have for some months past been making very careful tests of the Ljungström air preheater, which was fully described in our columns on July 7 last year, and as a result of these tests they have acquired an exclusive license for the system as applied to marine work, and a general license for land installations as well. The tests referred to were carried out on an ordinary Scotch marine boiler. The boiler was of the single-ended type, 11 feet long by 9 feet 6 inches diameter, with two Deighton corrugated furnaces 31⅞ inches internal diameter fitted with Howden patent furnace fronts. The boiler had 146 tubes, 2.5 inches diameter by 7 feet 6 inches long. The heating surface of the tubes was 716.13 square feet, of the furnaces 80.5 square feet and of the combustion chamber 113.37 square feet, making a total of 910 square feet. The grate area was 24 square feet, excluding the dead plate.

TABLE I.—*Summary of Test Figures Obtained from Cylindrical Type Marine Boiler, Fitted with Howden-Ljungström Air Preheater, at Works of James Howden and Co., Limited, Glasgow.*

Conditions.	Coal Burning.		Oil Burning.	
Date	31/1/23	14/2/23	5/3/23	6/3/23
Duration—hours	5	4	4	4
Steam pressure, lb. per sq. in.	164	167	161	161
Average feed temperature deg. F.	180	165	179	172
Class of fuel	Scotch Coal	Scotch Coal	Mexican Oil	Mexican Oil
Calorific value of fuel—B.Th.U.	13,055	12,300	17,574	17,574
Fuel burned per hour, lb.	540	636	453	495
Equivalent evaporated per lb. Fuel, lb.	11·12	10·19	15·97	15·76
Equivalent evaporation, per sq. ft. H.S., lb.	6·6	7·12	6·99	7·54
Temperature of flue gases entering air preheater, deg. F.	538	561	537	554
Exit temperature flue gases, deg. F.	217	231	195	215
Temperature air entering fan, deg. F.	76	66	77	77
Temperature of air leaving preheater, deg. F.	419	427	426	427
Temperature of air entering furnaces, deg. F.	405	415	398	409
Air pressure in reservoir around furnaces, in., W.G.	0·35	0·34	0·31	0·32
Air pressure at furnace door, in., W.G.	−0·06	0·02	0·16	0·15
Air pressure at ashpit door, in., W.G.	0·30	0·39	0·24	0·23
CO_2 in flue gases, per cent.	9·5	9·0	11·1	11·0
Thermal efficiency of boiler	84·36	85·49	87·8	87·13

Two furnaces, 31⅞ inches diameter, grate area 24 square feet. Total heating surface with coal, 910 square feet. Total heating surface with oil (bars removed), 1,036 square feet. Retarders in tubes.

The heating of the air supplied to the furnaces was effected by an air preheater of the Ljungström type, situated at the top of the uptake, and consisted of a cylindrical rotor, filled with a honeycomb arrangement of thin corrugated steel sheets, so arranged that the hot flue gases passed axially through one-half of the rotor, and the air for combustion passed in a contrary direction through the other half as the rotor slowly revolved. Two small fans on a common shaft propelled the air and gases along their respective paths. The total heating surface of the preheater was 5,110 square feet. There is in this type of apparatus no attempt at the transfer of heat from the flue gases to the incoming air through a metal wall. The rotor itself is heated by the gases, and the heat is swept off by the air for combustion.

All precautions were, of course, taken to ensure that tests intended to be comparative were carried out under as nearly identical conditions as possible, and they were supervised by engineers representing both parties concerned. Some of the results obtained are given in Table I.

It will be noted that on one test the temperature of the flue gases was reduced to the extraordinary low figure of 195 degrees F. Even if it were practicable to obtain equal results with an economizer, it would be very inadvisable to attempt such a reduction of temperature because of the condensation of moisture on the tubes. The efficiencies obtained will be recognized as quite extraordinary ones for any boiler, especially in view of the comparatively large amount of excess air which was passing through the fires.—"Engineering," June 29, 1923.

THE DEAERATION OF BOILER FEED WATER.

An Appliance for the Prevention of Corrosion of Boiler Tubes and Plates.

The treatment of water so as to render it suitable for use in steam boilers has for many years been an important item in the routine of the engineer-in-charge. In the past, the chief consideration has been the elimination of scale-forming matter from the water in order to prevent the formation of scale on the furnaces, tubes and shell plates. With modern plant, the amount of make-up feed water required has been reduced to a very small percentage of the total feed, and this make-up—when fresh water is used—is generally softened so that the amount of scale-forming matter contained in it is reduced to negligible quantities. The absence of scale is particularly noticeable in boilers supplied with feed make-up from evaporators, and the use of these is to be recommended because of the higher boiler efficiency obtained, owing to the fact that pure water is obtained from the evaporators.

With the disappearance of scale, however, the corrosion of the tubes and boiler plates tends to become more rapid. Searching investigation has proved that this corrosion is almost invariably due to the oxygen in the air which enters the boiler with the feed water. It can further be shown that substances generally considered corrosive, such as common salt, can be present in appreciable proportions without causing any corrosion, provided no free oxygen is present. The prevention of corrosion in boilers is, therefore, largely a question of the removal of air from the feed water. It is therefore of the greatest importance that steps should be taken to remove the air from the feed water. This may be accomplished in one of two ways—using either mechanical means or chemical means. The advantage of the mechanical method of deaeration lies in the fact that all gases in solution in the water are removed to the same extent in relation to their solubility, so that not only air but all other gases are eliminated.

The oxygen-content of water depends on its history prior to it being delivered to the boiler, but it may be taken at about 6 cubic centimeters to 10 cubic centimeters per liter. When water is boiled at a temperature of 212 degrees F. at atmospheric pressure, it can, when saturated, still contain 2 cubic centimeters of oxygen per liter. When the condensate is removed from a condenser by a separate pump, the quantity of oxygen depends on—

(*a*) The vacuum maintained in the condenser.
(*b*) The temperature of the water as extracted, and
(*c*) The partial air pressure maintained at the bottom of the condenser by the air-extracting pump.

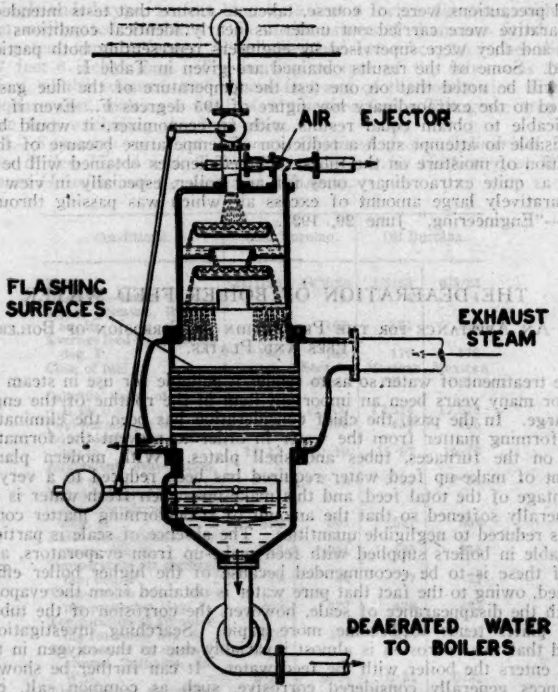


FIG. 1.—SECTION THROUGH THE DEAERATOR.

In a condensing plant the oxygen-content of the condensate may be taken as not exceeding 0.5 cubic centimeter per liter, but it will be less than this in plants maintained in an air-tight condition.

The question arises, to what extent can deaeration be carried out, and it may be stated that wherever possible, the oxygen-content should certainly be reduced to 0.1 cubic centimeter per liter, although in many cases it will be found possible to remove every measurable trace of oxygen. The prin-

ciple underlying the operation of the mechanical deaerator is as follows: When water is sprayed in a vessel containing steam, it becomes heated, and gives up a certain proportion of the contained air, but this process alone is insufficient for efficient deaeration. Water can, however, be completely deaerated by creating violent ebullition and foaming, and in the Contraflo type of deaerator this final effect is obtained by a flashing process, whereby the water is shattered into minute particles, following which the complete liberation of the contained gases occurs. This flashing can be accomplished by heating the water at a very low pressure and flashing it on a surface of high temperature (see Figs. 1, 2 and 3) or by first heating the water in a separate vessel and then introducing it into a chamber under vacuum. The former arrangement is preferable, because the entire apparatus can be made in the form of a self-contained unit.

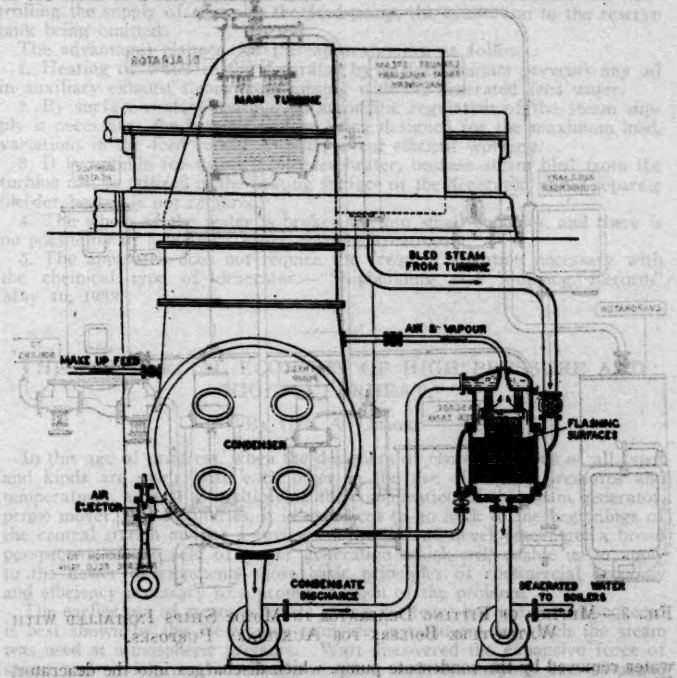


FIG. 2.—DEAERATOR OPERATED BY STEAM FROM LOW-PRESSURE TURBINE.

In the arrangement shown in Fig. 1, the lower part of the deaerator contains a bank of steam-heated tubes, over which the water falls. Preliminary deaeration takes place in the upper part of the deaerator, where the incoming water is heated to boiling point by the vapor rising from the hot tubular space below. This hot surface acts like a low-pressure flash boiler, and when the falling water, now at a temperature approaching boiling point,

comes into contact with the hot surface, it is repeatedly flashed as it falls from one tube to another on its way to the outlet, and the air originally contained in it is completely liberated. This air rises to the top of the vessel, and is withdrawn at the top by a small steam-operated ejector, which maintains a vacuum of from 10 inches to 20 inches in the deaerator. The float contained in the base of the deaerator controls the inlet of the water in accordance with the requirements of the boiler-feed pump. The steam required for operating the deaerator can, with very great advantage, be bled from the main turbine, and the cost of a separate heating unit is thus avoided. Such an arrangement is shown in Fig. 2, in which the steam required for the deaerator is bled from a low-pressure stage of the turbine. Make-up feed water can be admitted into the main condenser and the total

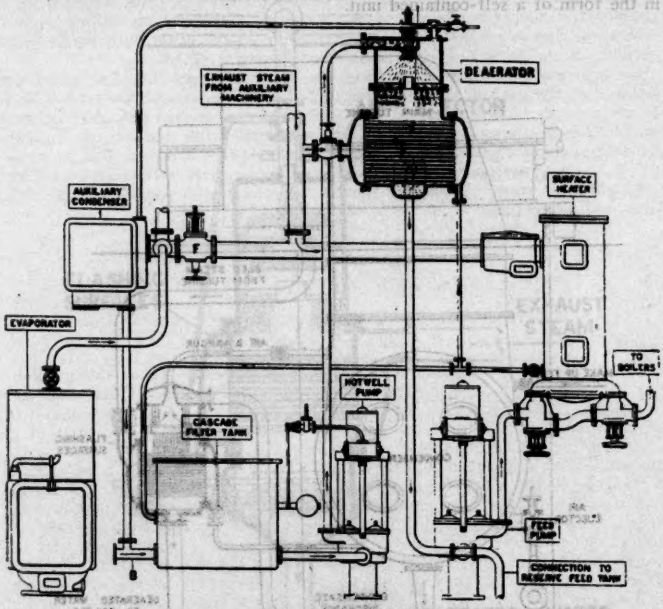


FIG. 3.—METHOD OF FITTING DEAERATOR IN MOTOR SHIPS INSTALLED WITH WATER-TUBE BOILERS FOR AUXILIARY PURPOSES.

water removed by the condensate pump, which discharges into the deaerator. The water from the deaerator, now practically free from air, may be delivered direct to the boiler or into a sealed storage tank.

This apparatus has been installed on a number of vessels during the past few years, and is particularly advantageous on ships fitted with water-tube boilers, where it is particularly necessary to remove the air from the feed-water in order to prevent corrosion of the boiler-tubes. Fig. 3 illustrates diagrammatically the installation on a number of oil-carrying ships driven by Diesel engines, fitted with water-tube boilers for generating the steam required for the auxiliary machinery and for heating purposes. In Fig. 3,

all the water, drains, etc., flow into the cascade filter tank, in which the oil is removed, the cleansed water passing to the hot-well pump, which delivers it to the deaerator. The deaerated water flows from the base of the deaerator to the feed pump, which discharges through the surface heater to the boilers. The feed-pump suction is also connected to the reserve feed-tank, so that when the quantity of water from the deaerator is greater than that required for boiler-feed purposes, the surplus water will flow into the reserve-feed tank and be available for use in the event of the supply of water from the deaerator being less than that required by the boilers. On ships where only a small quantity of auxiliary exhaust steam is available, the surface heater on the discharge side of the feed pump may be omitted, and the whole of the feed heating carried out in the deaerator. When a boiler-feed regulator is not fitted, a float may be provided in the base of the deaerator for controlling the supply of steam to the feed-pump, the connection to the reserve tank being omitted.

The advantages claimed for this apparatus are as follows:

1. Heating of water in the deaerator by surface contact prevents any oil in auxiliary exhaust steam from mixing with the deaerated feed water.

2. By surface-contact heating, no automatic regulation of the steam supply is necessary; the heating surface being designed for the maximum load, variations in the feed supply do not prevent efficient working.

3. It is suitable for use as a bleeder heater, because steam bled from the turbine can be utilized in the heating surface of the deaerator, and a separate bleeder heater is not required.

4. The whole of the water is broken up into small particles, and there is no possibility of a body of water escaping treatment.

5. The apparatus does not require the frequent attention necessary with the chemical type of deaerator.—"Shipbuilding and Shipping Record," May 10, 1923.

THE COMMERCIAL ECONOMY OF HIGH PRESSURE AND HIGH SUPERHEAT.*

By Geo. A. Orrok.

In this age of progress, when the designers of central stations of all types and kinds are vying with each other in the use of higher pressures and temperatures, as well as multiplying the complication of the steam generator, prime mover, and auxiliaries, it is expedient to go back to the beginnings of the central station and by a review of the line of development get a broad perspective of the field of power generation which will enable us to apply to the newer developments those basic principles of commercial economy and efficiency necessary to a proper solution of the problem.

The earlier use of steam when its expansive force has not been discovered is best shown by the Newcomen pumping installations in which the steam was used at atmospheric pressure. Watt discovered the expansive force of steam and so improved the boiler, engine, and steampiping that 15-pound gauge could be used economically. From his time until about 1900 rising pressure kept pace with improvements in materials and in the design and construction of steam generators and piping, with occasional excursions into the realms of higher pressure and temperature. Practice at this date (1900) may best be illustrated by the work of the Wildwood engine. Since 1900 improvements in steam generators, piping, and prime movers have been made with increasing rapidity, and many modern plants are running on

*Proceedings of the American Society of Mechanical Engineers.

250 pounds and 200 degrees of superheat, using standard piping and valves with the newer designs of boilers, superheaters, and turbines. Table I shows the pressures and economies theoretically possible and attained in a central station, with all the results reduced to British thermal units in the coal per kilowatt-hour of useful work.

Our engineers are now using better and more uniform materials than ever before. Our metallurgical and manufacturing methods have been greatly improved. Copper, wrought iron, and cast iron have been discarded for boiler purposes. Steel and even alloy steel are in common use, and their uniformity leaves little to be desired. Certain of the non-ferrous alloys are in common use and play their part in aiding the common security

TABLE I.

STEAM PRESSURES AND ECONOMIES THEORETICALLY POSSIBLE AND ACTUALLY ATTAINED IN CENTRAL STATIONS.

Year	Engine.	Pressure, Lb. per Sq. In.	Temperature, ° F.	Thermal Efficiency Carnot Cycle, %	Actual B.Th.U. per Kw.-hour.	Actual Over all Thermal Efficiency, %
1770	Newcomen	15	212	2·7	416,000	0·82
1810	Watt	30	300	9·17	61,000	5·6
1900	Wildwood	215	400	20·6	19,200	17·75
1922	Modern ..	265	650	35·00	18,000	19·00

which characterizes modern installations. Copper boilers and 212 degrees maximum temperature gave place to wrought-iron boilers and 30 to 40-pound pressure with a maximum temperature of 300 degrees. The use of steel as a boiler material raised the pressure to 200 pounds and the temperature to 400 degrees, while our modern steels allow pressures up to 400 pounds with a maximum temperature of 750 degrees. In certain distillation plants temperatures as high as 1,100 degrees have been obtained, but the life of the material is short.

Now it is well known that steel when heated to comparatively low temperatures, say, 900 degrees or 1,000 degrees, loses its strength and becomes unfit to sustain loads, and the heat strains from even moderate heating of 300 degrees or 400 degrees may cause certain deformations of a highly unsatisfactory character. Cast-steel valves have deformed at 750 degrees to such an extent as to render them useless, and the "growth" of ordinary cast iron at temperatures above 450 degrees is known to be destructive; but notwithstanding this the later constructions at 250–300 pounds pressure and 700 degrees maximum temperature are commercial. The first cost is not excessive, repairs are moderate, and the life of the installation is all that can be desired.

COMMERCIAL LIMITS OF PRESSURE AND SUPERHEAT IN THE CENTRAL STATION.

What, then, are the commercial limits of pressure and superheat in the central station? In Fig. 1 we have calculated and plotted the theoretical thermal efficiency of the Carnot cycle, the Rankine cycle, and a regenerative cycle, using both saturated and superheat values for ranges up to 1,200 pounds and superheats of 750 degrees and above. It will be seen that the chosen regenerative cycle for saturation follows very closely the efficiency of the Carnot cycle, while the Rankine cycle falls below the Carnot cycle increasingly with rise of pressures. The superheat lines for both cycles are nearly parallel and maintain this characteristic over a wide range. The

gain from regeneration increases with pressure and is constant for equal superheat. For any final temperature the gain from pressure increments decreases, but the gain from regeneration increases with pressure increments. Thus at 750 degrees final temperature the Rankine efficiency increases from 33.5 to 40.5 per cent with pressure rise from 200 to 1,200 pounds, while the regenerative efficiency increases from 37 to 48.5 per cent with the same pressure increase.

Fig. 1, dealing with theoretical efficiencies only, does not show the station losses, which aggregate around 45 per cent. Grouping the losses, we may

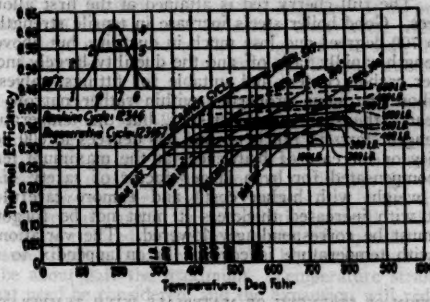


FIG. 1.

say that the generator efficiency is 97 per cent, turbine efficiency 78 per cent, boiler efficiency 80 per cent, leaving for auxiliaries, piping, and radiation 91 per cent, all of which multiplied together give 55 per cent efficiency.

TABLE II.

SAVINGS EFFECTED THROUGH INCREASE IN STEAM PRESSURE.

Case.	Cycle.	Pressure, Lb.	Max. Temp. ° F.	Theoretical Efficiency, %.	Station Efficiency, %.	Actual Efficiency, %.	B.Th.U. per Kw.-hour.
1	Rankine ...	200	750	33·5	55	18·4	18,500
2	Rankine ...	1200	750	40·5	55	22·3	15,300
	Saving ...	—	—	21	—	—	3,200
3	Regenerative	200	750	37	55	20·35	16,800
4	Regenerative	1200	750	48·5	55	26·7	12,800
	Saving ...	—	—	33·3	—	—	4,000

Table II shows how these figures work out for the above suppositions. The saving by regeneration when using 200-pound pressure (Case 1—Case 3) is 1700 B.T.U., and when using 1,200-pound pressure (Case 2—Case 4) is 2,500 B.T.U. Considering the trend of the curves in Fig. 1 on this page, comparatively little is gained by further superheating, but there has been no attempt to go above 800 degrees, and the flattening out is much more marked as the temperature increases.

Water has its critical temperature at 704 degrees, and the critical pressure is about 3,200 pounds. Pressures up to this limit may therefore be considered, and since oil-still temperatures up to 1,100 degrees have been used, we may consider superheat temperatures up to that point. We know that the first allotropic change in steel occurs at about 1,300 degrees, and 1,100 degrees is well below this point. Here, however, we must look into the physical properties of the steels used in power-plant construction.

When steel is heated the first sign of change—a dull red just visible in the dark—appears at about 750 degrees. Visibility in daylight begins at about 850 degrees to 900 degrees, and a full dark red is attained at about 1,100 degrees. The full cherry red is attained at the first allotropic change of 1,300 degrees. Good boiler steels increase in tensile strength up to about 600 degrees to 700 degrees and lose much in ductility, but above 800 degrees the tensile strength rapidly falls off and the ductility largely increases. Cast steels of proper carbon content and suitable for fittings, valves, and turbine construction show the same properties within rather narrow limits. Pipe steels have nearly the same characteristics, but the temperature at which the tensile strength starts to fall is around 550 degrees to 600 degrees. This lack of ductility or increase of brittleness at the maximum temperature of use must be compensated for by a larger factor of safety (*i. e.*, increased thickness and weight with higher cost) and a more careful selection of material. But with increased thickness it must not be forgotten that the safe stresses must be correspondingly lowered. The variation of strength of materials with temperature is covered in an appendix to the complete paper.

VARIATION OF STRENGTH OF MATERIALS WITH TEMPERATURE.

This subject may be considered from five points of view, depending on the conditions under which the materials are to be used.

(*a*) *Boiler Material.*—Here we know that the maximum temperature cannot exceed 704 degrees, the critical temperature of water, and that the fire side of the tube or drum can only be a few degrees hotter. If 1,200 to 1,800 pounds per square inch be the chosen pressure, the water temperature is 625 degrees at maximum and the fire side will not exceed 725 normally. This temperature is well within the maximum strength zone, and only care is needed to secure sufficient ductility.

(*b*) *Superheater Material.*—The maximum temperature of the inner surface of the tubes may be a little higher than the maximum superheat. The outer temperature depends on the position in the boiler. Radiant-heat superheaters are now commercial where the pressure parts are protected with a heat-absorbing and conducting covering of cast iron. The fire surfaces may be 1,400 degrees to 1,600 degrees, while the inner-wall temperature is as low as 700 degrees, but 1,100 degrees is comparably easy of attainment. Bare-tube superheaters would have a maximum outside temperature of around 1,200 degrees, but the life at this temperature is unknown. The maximum safe point may be taken as 800 degrees superheat temperature and 950 degrees outside temperature.

Neither the tube manufacturers nor superheater manufacturers are afraid of these conditions, and I believe they are prepared to guarantee their product under such conditions.

(*c*) *Piping Material.*—Under this category the condition is markedly different. The steam side of the pipe is below the maximum temperature of the steam. The outer surface is much lower, and the thickness of the material can be increased to compensate for the lower elastic limit of the material. The ductility is improving with temperature increase. Pipe joints are the only serious trouble, and the modified Van Stone joint with the pipe edges welded for tightness is the apparent solution. Experience with this joint has been satisfactory.

(*d*) *Valves and Fittings.*—Valves and fittings must necessarily be made of castings, and the low-carbon open-hearth steel used, while falling off in strength sooner than the worked material used in boiler and piping, has still a respectable elastic limit at 1,100 degrees. As in the last category, the inside-wall temperature is lower than the steam temperature, with still lower temperatures at the outer wall. But the shape of the casting is all-important. Globe valves of the double-beat or Wanick type can be made with practically no flat surfaces and with two axes of symmetry. Internal pressures do not seriously deform this design, but external strains may cause minor troubles. Throttle valves and marine stop-valves are usually of this type, and can be kept tight at 750 degrees. It is probable that a temperature of 1,100 degrees could be safely undertaken with these valves if special precautions were used in the design of the seat, disc, and stem. Flexible-disc, double-beat valves are used in Europe at temperatures in excess of 750 degrees.

Gate valves have only one axis of symmetry and flat surfaces of considerable size. Here seat and disc troubles are encountered at temperatures above 700 degrees, and it may be some time before the manufacturers can guarantee a tight job at the higher temperatures. This type of valve is peculiarly sensitive to outside strains tending to deform the seats and discs.

(*e*) *Turbine High-pressure Ends.*—Pressures can make little difference here as the high-pressure parts are comparatively small, and without doubt the pressure will be reduced 50 per cent in the first nozzle. There need be no high-pressure stuffing-box, so that the highest pressure to be packed against will be about half the maximum. Temperature strains will be the only thing that need be considered, and the expansion will reduce the superheat in the first-stage nozzle to a workable value, which is 750–850 degrees. It is to be noted that steamchest or nozzle-box troubles have been the only prominent troubles where 400-pound and 750 degrees superheat have been tried out, and it thus appears that a proper design will obviate most of the trouble. Cast or forged steel must be the material used.

In general, the turbine designers say that 1,200–1,500-pound pressure has no terrors for them, and can be used when desired. They can supply the apparatus. Temperatures are more troublesome, and no one apparently cares to go much above 700–750 degrees at the present time. Superheater and valve materials at present seem to be the deciding factor in the use of high pressures and temperatures, and there is apparently very little which careful design and the selection of proper materials will not overcome.

Only a portion of the central-station installation is affected by high pressures and temperatures, especially where electric auxiliaries are used. The boiler and piping system and prime mover cover the entire list, and in the usual central station this has represented about 40 per cent of the entire installation cost. Latterly, with the larger station the percentage is around 30, and may be taken at 25 per cent for the newer and larger stations of which Hell Gate and Calumet are types. If the station cost be taken at 100 dollars per kilowatt as an average figure this portion of the installation has cost 25 dollars, and at 15 per cent the fixed charges are 3.75 dollars per year, or on 5,000 hours' use 0.07 cent per kilowatt-hour. What, then, will be the extra cost of this apparatus when designed for higher pressures and temperatures? Standard boilers today can be bought up to 275 pounds pressures; 300 pounds necessitates thicker plates and tubes, and prices advance accordingly. Four-hundred-pound boilers have been purchased recently, and 500-pound boilers have been built. Flash boilers of the Serpollet and De Laval types, using pressures in excess of 1,200 pounds and coil boilers of the Herreshoff type using equally high pressures, have been built as experiments or in toy sizes as for automobile work or for high-speed launches. Schmidt's water-tube boiler working at 900 pounds has been run

14,000 hours with marked success. Boilers for 400-pound pressure have been offered at about 50 per cent increase in price, and 1,200-boilers have been figured at a 100 per cent increase in price.

Turbine designers are at work on designs arranged to utilize higher pressures, but higher temperatures seem more troublesome to them. I have no estimate, but we can safely apply the same percentage increases of price to the turbine that obtain with the boiler. Steampiping decreases in size with increasing pressure, so much so that it is doubtful if a 1,200-pound steam line would cost more than a 200-pound line of the same capacity. The volume of high-pressure steam at 1,100 degrees maximum temperature is quite uncertain, but the chances are that our figures are not more than 10 per cent from the truth.

SUMMARY.

We may then summarize as follows:

(*a*) Pressures up to 1,200–1,500 pounds at least are commercial and may be attained without serious difficulty.

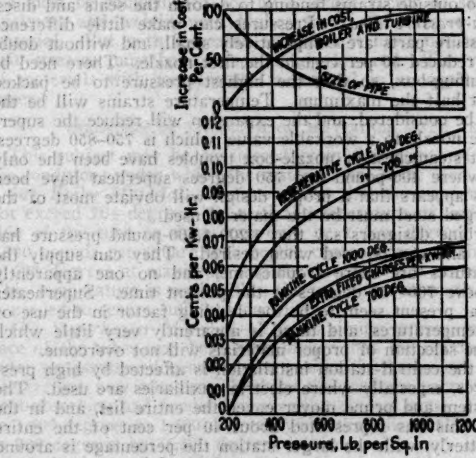


FIG. 2.—COMMERCIAL ECONOMY.

(*b*) Temperatures for the present are commercial up to 700–750 degrees, which should not be exceeded until our materials for valves and superheaters are improved. More extended experience is necessary before the range of 800 degrees and over is attempted in a commercial installation.

(*c*) With the completion of the Steam Table Research it will be possible to calculate accurately just where the possible limits of economy in pressure and temperatures may be located. At the present time the uncertainty of all values above 200 pounds pressure and 4,000 degrees F. temperature render most of our calculations of only academic value.

(*d*) There still appears to be more economy in a closer study of operation and construction losses—resulting in improving the efficiency of central stations installed in accordance with the present accepted canons of design—

TABLE III.—SAVINGS IN COAL.

Pressure Lb. per Sq. In.	Superheat Temp., ° F.	Carnot Efficiency.	Regen. Efficiency.	Rankine Efficiency.	Regen. × 0·55	Rankine × 0·55	Regen. B.Th.U.	Rankine B.Th.U.	Saving Over R. 200lb. 700° B.Th.U. per Kw.-hour.		Saving in Lb. Coal per Kw.-hour (13800 B.Th.U.).	
									Regen. C.	Rankine C.	Regen. C.	Rankine C.
200	700	53·5	37	33·4	20·3	18·35	16,820	18,620	1800	—	0·13	—
400	700	53·5	41·2	36·3	22·6	19·95	15,100	17,120	3520	1500	0·255	0·109
600	700	53·5	43·5	38	12·9	20·9	14,300	16,350	4320	2270	0·313	0·1645
1200	700	53·5	47·5	40	26·1	22	13,100	15,500	5520	3120	0·40	0·226
200	1000	63	38	35	20·9	19·25	16,350	17,750	2270	870	0·165	0·065
400	1000	63	42·8	37·6	23·5	20·7	14,500	16,500	4120	2120	0·299	0·1535
600	1000	63	45	39·5	24·7	21·7	13,820	15,720	4800	2900	0·348	0·21
1200	1000	63	49	41·5	26·9	22·8	12,700	14,970	5920	3850	0·43	0·257

than in the attempt to widely increase the temperature range of our heat cycles. While we may save 400 B.T.U. per kilowatt-hour by using a regenerative cycle and increasing pressure and temperature, it should be possible to save nearly as much by a reduction to the lowest limits of the heat losses which we know exist in our present installations.

SAVINGS OF REGENERATIVE CYCLE.

In Table III we have given for 700 degrees and 1,000 degrees maximum temperature and for four pressures the theoretical efficiency of the three cycles used, the practical efficiency of the Rankine and regenerative cycles, and the savings per kilowatt-hour for all conditions over the Rankine cycle at 200 pounds and 700 degrees maximum temperature in B.T.U. and pound of coal of 13,800 B.T.U. content.

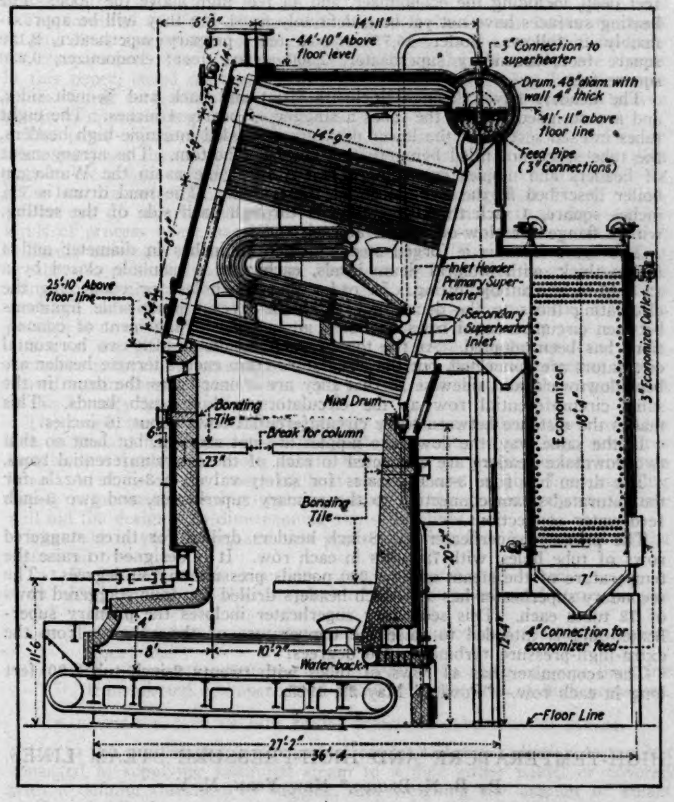
In Fig. 2 these savings are plotted in cents per kilowatt-hour on the basis of 6 dollars coal, together with the extra-fixed-charges curve. At the top of the figure are two curves showing the percentage increase in cost of boiler and turbine and the decrease in size of pipe required for a given quantity of steam. It will be seen that the extra-fixed charges are always greater than the coal saving for the Rankine cycle (ordinary operation) at 700 degrees maximum temperature, and that the additional gain for the 1,000 degrees maximum temperature barely exceeds the additional fixed charges. We cannot then expect much from increasing pressures and the ordinary methods of using steam. Increasing temperatures help to some extent, but not enough to be attractive, since the saving is nearly all used up by the additional fixed charges.

With the regenerative cycle the results are much better, and useful savings are indicated, increasing considerably with pressure and slightly with temperature. It is evident that some such regenerative cycle must be used, but the rewards from increasing temperature are not commensurate with the costs. Many regenerative cycles are possible, but the one used shows nearly the maximum economy. Such cycles need many heat interchangers and a number of separate feed-pumps for the close approach to theoretical figures. Indeed, Thurston and Stanwood in 1899 pointed out that with an infinity of heaters and pumps the Carnot efficiency might be equalled. So far, however, four or five stages have been the ultimate practical application, and not more than three have stood the commercial test. It would appear that considerably better design must be put in the apparatus if many steps are proposed, and also that more consideration must be given to the operating difficulties.—"Mechanical World," May 25, 1923.

CALUMET STATION TO HAVE 1,200-POUND BOILER.

As another step in the progress toward the use of higher steam pressures, it is interesting to learn that about the first of the year the Commonwealth Edison Company, of Chicago, ordered one boiler designed for an operating pressure of 1,200 pounds per square inch. It will be an experimental installation, and the intention is to use this boiler at the Calumet Station and to put in with it a small "extra-high-pressure" turbine taking steam from the boiler at 1,200 pounds pressure and exhausting at 300 pounds pressure into the steam mains which supply the large turbines of the station. The turbine has not yet been ordered and the details of the system have not been completely worked out. The details of the boiler design, however, have been practically completed by the Babcock & Wilcox Co. in conjunction with the consulting engineers of the Commonwealth Edison Company.

The unit is of the inclined-header, cross-drum type and comprises a lower deck of eight-high sections of 2-inch tubes having a setting height of 25 feet 9¾ inches, a primary and a secondary superheater in an interdeck space



SECTION THROUGH BOILER, SHOWING SECONDARY SUPERHEATER AND FOUR-INCH THICK DRUM.

8 feet 1¼ inches between decks, an upper deck of seventeen-high sections of 2-inch tubes, horizontal 3¼-inch circulating tubes entering a 48-inch cross-drum, and a Babcock & Wilcox steel contraflow economizer. The lower deck is not baffled; the upper deck has a vertical baffle, causing the gases to make two passes. The complete unit is about 28 feet wide, 36½ feet deep, including the economizer, and 45 feet high above the floor. The heating surfaces have not yet been definitely fixed, but they will be approximately as follows: Boiler, 15,750 square feet; primary superheater, 2,120 square feet; secondary superheater, 3,300 square feet; economizer, 9,230 square feet.

The headers have 1¼-inch thickness front and back and ⅝-inch sides, and are designed to give the tubes a stagger of nearly 4 inches. The eight tubes in each section of the lower deck are expanded into nine-high headers, the tube space not filled being that next to the bottom. The arrangement of headers and nipples connecting them is the same as in the Waukegan boiler described in the September 12, 1922, issue. The mud drum is 7¼ inches square, 1 inch thick and extends through each side of the setting, with a flange for blow-off valves at each end.

The cross-drum is a forged steel cylinder 48 inches in diameter and 4 inches thick, with integral drum heads, each with a manhole closed by a 12 × 16-inch manhole fitting. In order that the holes for connecting the circulating tubes to the drum may leave the largest practicable ligaments between circumferential rows of holes, an unusual arrangement of connections has been adopted. At the top of each uptake header two horizontal circulators are connected, but the circulators from each alternate header are bent downward and sidewise so that they are connected to the drum in the same circumferential row as the circulators without such bends. This makes the distance between these circumferential rows about 16 inches.

In the same way, the downtake nipples are not straight, but bent so that two downtake headers are connected to each of these circumferential rows.

The drum has four 3-inch nozzles for safety valves, a 3-inch nozzle for the saturated-steam connection to the primary superheater, and two 3-inch feed-water connections.

The primary superheater has 8-inch headers drilled for three staggered rows of tube holes, with 72 tubes in each row. It is designed to raise the temperature of the steam under 1,200 pounds pressure to 750 degrees. The secondary superheater has 11¾-inch headers drilled for four staggered rows of 72 tubes each. This secondary superheater incloses the primary superheater and is intended to raise the temperature of the exhaust from the extra-high-pressure turbine to 750 degrees.

The economizer has 44 rows of tubes with twenty 2-inch tubes 20 feet long in each row.—"Power," May 23, 1923.

HIGH-TEMPERATURE AND HIGH-PRESSURE STEAM LINES.

By B. N. Broido,* New York, N. Y.

The pronounced tendency of modern power-plant designers to use high superheat and high pressure, as well as the recent extensive use of highly superheated steam in process work, lead the author to make a review of the available data and formulas on radiation and friction losses in pipe lines. All tests to determine the radiation losses of pipe lines carrying high-temperature steam made in this country have been practically exclusively

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with electrically heated pipes. This paper gives results and an analysis of tests made with superheated steam flowing through a pipe line. The heat-transmission coefficients of bare and covered pipes at various steam velocities, superheats, and pressures are given. Suggestions are made regarding the steam velocities to be selected in order to transmit steam more economically.

The formulas available at present for figuring the friction losses of steam in pipe lines are limited to the range for which they were established, or for comparatively low pressures. Corrections of these formulas are shown in this paper, based on observations of a pipe line carrying high-pressure steam, which corrections may be used by designers of high-pressure power plants in dimensioning their pipe mains.

The paper further treats of pipe lines carrying superheated steam for other purposes than power. Recommendations as to the velocities or pipe sizes to be chosen are given, as well as a few examples from practice to show how the velocities selected were advantageously applied.

The use of superheated steam in power plants, as well as for various kinds of process work, has become so extensive during recent years that it is hardly necessary to enlarge on its advantages. The fuel saving effected by the use of superheated steam, the lower maintenance cost of turbines using it, and the numerous advantages it has in various kinds of process work are some of the factors which have helped to assign to it the important position it now occupies in the opinion of technical men.

One of the phases in which engineers, power-plant operators, and managers are naturally interested in this connection is that of conducting superheated steam through pipe lines. Very little, however, has been published in this country on the subject.

The recent tendency in designing power plants, particularly those of large size, is toward high pressures. The high temperatures and high pressures involve new problems, among others being that of proportioning the pipe lines. A study of the losses of heat and pressure in pipe lines carrying highly superheated and high-pressure steam at the present opportune time will aid the designer in dimensioning the steam lines.

The field is very large, and this paper will therefore be confined to the discussion of—

(*a*) Radiation losses in pipe lines carrying superheated steam.
(*b*) Resistance to the flow or pressure drop of superheated steam in pipe lines.
(*c*) The most advantageous steam velocity, particularly for high-pressure lines; and
(*d*) Other features to be taken in consideration in designing pipe lines for transmission of superheated steam of either low or high pressures.

RADIATION LOSSES IN PIPE LINES CARRYING SUPERHEATED STEAM.

Up to the present the usual method of determining radiation losses has consisted in supplying saturated steam to a pipe either naked or covered with a definite thickness of lagging, and weighing the amount of steam condensed. Such experiments, however, are mostly inaccurate, due to the fact that no separator completely removes the moisture from the steam, and also because of the difficulties in completely draining the tested pipe. So far as the author is aware, no experiments with superheated steam in pipe lines have been made in this country.

The author has had the opportunity to study this subject and to examine the results of tests conducted in Europe in which both saturated and superheated steam were used, particularly the tests made by Dr. Bernard and Herr Eberle of Magdeburg and Munich, respectively.* Table 1 shows the

*Mitteilungen über Forschungsarbeiten (Berlin), V. D. I., Hefte 21 and 78.

results of tests with saturated steam in bare pipes of approximately 3-inch and 6-inch internal diameter. The surfaces given in the table include the surface of the flanges. It is of interest to note that while the temperature of the pipe wall was very near to that of the steam, the temperature of the flanges was slightly lower. The difference in some cases is as high as 30 degrees F.

TABLE 1. HEAT LOSS FROM 3-IN. AND 6-IN. BARE PIPES—SATURATED STEAM.

Pipe Line								Heat Loss	
Length, ft.	Inside diam., in.	No. Flanges	Pressure abs., lb. per sq. in.	Steam temp., deg. fahr.	Air temp., deg. fahr.	Temp. diff. betw. steam-air, deg. fahr.	Condensed water for 1 ft. length, lb. per hr.	Per sq. ft. surface per hr., B.t.u.	Per sq. ft. per deg. temp. diff. per hr., B.t.u.
87.25	2.76	6	46.60	274.46	60.98	213.48	0.5174	575.7	2.672
			46.01	273.56	59.54	214.02	0.5168	575.3	2.672
			46.45	274.10	61.88	212.23	0.5114	570.2	2.672
			96.43	322.52	65.66	256.86	0.6945	744.0	2.876
			96.87	322.88	62.96	259.92	0.7063	755.9	2.876
			95.84	322.16	70.34	251.82	0.6982	758.5	2.958
			190.22	374.72	76.10	298.62	0.9092	932.4	3.100
			191.69	375.26	71.00	304.26	0.9415	964.7	3.142
85.28	5.91	7	47.78	275.9	72.50	203.40	1.0362	545.4	2.672
			48.95	277.34	78.26	199.08	1.0087	537.2	2.652
			99.23	321.68	89.06	235.62	1.3628	689.3	2.897
			98.78	324.32	86.36	237.96	1.3808	698.9	2.917
			192.13	375.44	87.08	288.36	1.9320	935.7	3.223
			191.98	375.44	95.75	279.69	1.8715	906.9	3.223

The tests were also conducted with different types of insulation, the results, however, being similar to those obtained in like tests in this country.

Tests to determine the heat loss by transmission of superheated steam can be carried out only with steam that is flowing. The heat content of steam, both at the inlet and outlet of the pipe, must be determined, and the difference multiplied by the amount of steam flowing through the pipe gives the heat loss. Superheated steam flowing through a pipe may change its heat content—

(*a*) By decreasing its temperature.
(*b*) By pressure drop.
(*c*) By partial condensation.

The accurate measurement of any condensed water would be impossible, due to the fact already mentioned that separators are not reliable. To be accurate, tests with superheated steam must therefore be conducted with sufficiently high temperatures so that the temperature of the pipe wall at the outlet does not descend below that of saturated steam at the same pressure.

Knowledge of the wall temperature was considered important, not only to determine the minimum superheat required to eliminate condensation, but also to determine the coefficient of heat transfer from superheated steam to a pipe wall.

In view of the fact that the velocity of superheated steam doubtless has an influence upon the heat transmission, tests were conducted with different velocities. The results of tests conducted with bare pipes were highly impressive. At a velocity of about 100 feet per second and steam temperatures of 291 to 390 degrees C. (556 to 734 degrees F., corresponding to 221 and 399 degrees superheat), the temperature difference between the steam and pipe wall was 34 degrees C., while a_1 the heat-transmission coefficient was 35. With steam velocities of 33 feet per second the difference between the steam and the pipe-wall temperatures was greater. At the lower steam temperatures of 220 to 267 degrees C. (428 to 513 degrees F.) the temperature difference was 45 to 65 degrees C. (86 to 122 degrees F.) and the

TABLE 2. COEFFICIENTS OF HEAT-TRANSMISSION FOR BARE AND COVERED PIPES AT VARIOUS STEAM TEMPERATURES AND SAVINGS RESULTING FROM COVERING PIPE—SUPERHEATED STEAM

Final steam temp., deg. fahr.	3-in. Line			6-in. Line		
	K		Saving through covering, per cent	K		Saving through covering, per cent
	Bare pipe, B.t.u.	Covered pipe, B.t.u.		Bare pipe, B.t.u.	Covered pipe, B.t.u.	
212	2.4	0.47	80.4	2.32	0.381	83.6
230	2.48	0.47	81.1	2.42	0.383	84.2
248	2.57	0.47	81.7	2.52	0.385	84.7
266	2.64	0.47	82.3	2.62	0.387	85.3
284	2.73	0.47	82.8	2.72	0.389	85.7
302	2.81	0.47	83.4	2.81	0.391	86.1
320	2.89	0.47	83.8	2.91	0.393	86.5
338	2.98	0.47	84.2	3.01	0.395	86.9
356	3.06	0.47	84.6	3.1	0.397	87.2
374	3.14	0.47	85.0	3.2	0.399	87.5
392	3.22	0.47	85.4	3.3	0.401	87.9

heat-transmission coefficient a_1 was 16. By decreasing the velocity of the steam one-third, the coefficient of heat transmission between superheated steam and the pipe wall was decreased to one-half of the first value. This indicates clearly that the behavior of superheated steam is different from that of saturated steam as far as imparting heat to a metallic wall is concerned, which is due to the fact that saturated steam, in transferring heat to the pipe wall, partly condenses or gives up some of its latent heat, which heat is not as closely combined with the steam as the sensible heat of a liquid or the heat of superheat. This fact often is not understood, even by technical men.

While the results of the tests with bare pipe—particularly the variation of the heat-transmission coefficient at different temperatures and velocities—are not important in connection with pipe lines which invariably are insulated, they are of considerable interest as far as the use of superheated steam for heating and drying purposes is concerned.

The experiments demonstrate that the heat-transmission coefficient increases considerably with the increased velocity, and also that the wall temperature depends not only upon the steam temperature but also upon its velocity.

Table 3 gives the results obtained with superheated steam in covered pipes. The same covering was used as for the test with saturated steam. A comparison of the values of Tables 2 and 3 for saturated and superheated steam in covered pipes brings out the fact that for the steam-temperature range from 100 to 200 degrees C. (212 to 392 degrees F.), with either saturated or superheated steam, the heat-transmission coefficient K remains practically the same. With increased temperatures this coefficient shows a tendency to rise considerably. For temperatures from 100 to 200 degrees C. (212 to 392 degrees F.), K for lines with pipes and flanges covered is about 0.47. It increases, however, with the steam temperature, up to 0.69 at 400 degrees C. (752 degrees F.).

TABLE 3 VARIATION OF HEAT LOSSES, HEAT-TRANSMISSION COEFFICIENTS FOR BARE AND COVERED PIPES AND SAVINGS RESULTING FROM COVERING PIPE, WITH TEMPERATURE DIFFERENCE BETWEEN STEAM AND AIR —SUPERHEATED STEAM

Temperature difference between steam and air, deg. fahr.	Heat loss per square foot of outside pipe surface per hour		Saving through the covering, per cent	K	
	Bare pipe, B.t.u.	Covered pipe, B.t.u.		Bare pipe, B.t.u.	Covered pipe, B.t.u.
212	453	78.5	82.7	2.50	0.43
257	606	103.2	83.0	2.67	0.45
302	773	130.2	83.2	2.83	0.48
347	971	160.5	83.5	3.06	0.51
392	1184	192.4	83.7	3.26	0.53
437	1394	225.2	83.8	3.43	0.55
482	1637	260.8	84.1	3.63	0.57
527	1909	303.4	84.1	3.83	0.60
572	2192	344.1	84.3	4.04	0.63
617	2497	384.8	84.6	4.22	0.65
662	2804	429.2	84.7	4.43	0.67
707	3135	478.7	84.7	4.61	0.70
752	3149	529.8	84.8	4.81	0.73

While from the tests one would conclude that the direct radiation losses for superheated steam are not appreciably lower than those of saturated steam of the same temperature, there are, however, a few other points to be considered in estimating the relative advantages of the two so far as their transmission through pipe lines is concerned. With saturated steam any heat radiated causes a part of the steam to be condensed, and the condensate, particularly for long pipe lines, is not returned to the boiler but is discharged through traps and wasted, so that in addition to the direct heat lost by radiation there is a further loss of the liquid heat of the condensate, which in some cases, particularly for high-pressure steam, may amount to 25 per cent of the radiated heat. With superheated steam the heat lost only decreases the temperature and no condensation occurs as long as the steam remains superheated so that no additional heat besides that radiated is wasted.

Another important fact to be considered in connection with superheated steam is that a considerably higher velocity in the pipe lines is permissible.

TABLE 4. COMPARISON OF CAPACITY, RADIATION LOSS AND PRESSURE DROP OF PIPE LINES TRANSMITTING SATURATED AND SUPERHEATED STEAM

Saturated Steam								Superheated Steam							
4-in. Pipe (Extra Strong, Above 125 Lb. Pressure)															
Press. abs., lb. per sq. in.	Press. drop, lb. per sq. in.	Velocity, ft. per min.	Wt. of steam, lb. per hr.	Million B.t.u. trans. per hr.	Rad. loss	Total loss	Per Cent loss	Press. abs., lbs. per sq. in.	Deg. superheat	Press. drop, lb. per sq. in.	Velocity, ft. per min.	Wt. of steam, lb. per hr.	Million B.t.u. trans. per hr.	Rad. loss	Per cent loss	Per cent saving
115	0.52	3000	4,000	4.18	10,000	13,400	0.47	115	100	1.31	6,000	7,000	7.40	20,000	0.34	37.7
									150	1.13	6,000	6,870	7.15	20,000	0.40	16.8
									200	1.12	6,000	6,800	6.92	20,000	0.43	.....
165	0.70	3000	5,170	5.35	17,000	20,180	0.42	165	100	1.75	6,000	6,575	6.68	20,000	0.36	38.3
									150	1.57	6,000	6,400	6.22	21,200	0.24	19.01
									200	1.38	6,000	7,810	8.00	25,000	0.48	-4.76
215	1.43	3000	6,660	6.87	19,000	24,375	0.37	215	100	2.17	6,000	11,000	12.40	32,000	0.32	37.9
									150	2.04	6,000	10,720	11.75	33,400	0.25	24.3
									200	1.94	6,000	10,120	11.30	30,000	0.64	8.1
4-in Pipe (Extra Strong, Above 125 Lb. Pressure)															
115	0.30	6000	8,000	8.35	22,000	26,700	0.28	115	100	0.75	6,000	10,000	11.01	27,000	0.21	37.6
									150	0.71	6,000	13,000	14.25	42,000	0.26	18.6
									200	0.67	6,000	14,710	16.25	40,000	0.20	.....
165	0.46	6000	10,000	10.78	30,000	35,870	0.28	165	100	1.04	6,000	10,200	11.40	30,000	0.11	60.8
									150	0.99	6,000	10,000	11.01	40,000	0.20	28.7
									200	0.94	6,000	17,000	18.15	52,000	0.26	13.3
215	0.78	6000	12,000	12.75	37,000	39,000	0.26	215	100	1.29	6,000	22,000	20.09	42,000	0.15	42.36
									150							
									200							

115	0.35	3000	3,000	6.75	20,250	20,700	0.29	115	100	0.71	6,000	13,000	16.25	29,250	0.25	19.6
									125	0.47	6,000	14,730	16.25	29,000	0.20	.....
165	0.46	3000	10,350	10.78	32,350	33,570	0.29	165	150	1.33	6,000	20,280	21.40	38,480	0.18	40.6
									100	0.68	6,000	10,000	20.25	40,000	0.20	28.7
									125	0.94	6,000	17,000	20.15	32,030	0.26	13.8
215	0.72	3000	13,500	16.75	37,500	38,000	0.26	215	150	1.33	6,000	22,000	20.09	42,600	0.15	42.35
									150	1.32	6,000	24,320	24.25	45,000	0.18	20.8
									200	1.18	6,000	25,000	25.00	50,700	0.22	15.4

8-in. Pipe (Extra Strong Above 125 Lb. Pressure)

115	0.35	3000	10,500	13.0	31,500	32,500	0.21	115	100	0.71	7,000	22,410	24.5	49,000	0.14	23.6
									125	0.47	7,000	20,300	23.1	45,100	0.17	10.06
									200	0.94	7,000	25,710	31.25	63,700	0.20	4.76
165	0.46	3000	20,370	24.55	32,000	42,000	0.17	165	150	0.80	7,000	31,470	44.45	91,000	0.12	36.4
									200	0.94	7,000	32,510	42.6	83,770	0.14	16.02
									100	0.68	7,000	28,500	41.66	83,510	0.17	.....
215	0.70	3000	31,150	31.7	35,500	46,500	0.15	215	150	1.32	7,000	35,040	47.25	95,120	0.10	39.8
									150	1.32	7,000	29,600	35.00	69,740	0.12	20.0
									200	1.18	7,000	40,000	48.97	79,000	0.24	6.67

10-in. Pipe (Extra Strong Above 125 Lb. Pressure)

115	0.36	4000	20,000	24.75	32,410	40,700	0.14	115	100	0.68	8,000	30,300	41.67	83,700	0.10	23.5
									125	0.65	8,000	24,200	29.20	68,000	0.12	10.26
									200	0.93	8,000	31,200	37.16	78,530	0.14	.....
165	0.46	4000	44,250	45.85	43,000	52,100	0.10	165	100	0.91	8,000	77,300	93.75	93,670	0.08	23.3
									125	0.91	8,000	72,000	79.00	74,000	0.09	26.3
									200	0.86	8,000	40,000	78.80	83,020	0.10	6.66
215	0.72	4000	56,200	60.30	46,000	56,000	0.16	215	100	1.18	8,000	70,100	108.9	68,000	0.06	40.0
									150	1.15	8,000	60,000	108.5	72,000	0.08	20.0
									200	1.07	8,000	57,000	86.9	92,000	0.06	10.0

12-in. Pipe (Extra Strong Above 125 Lb. Pressure)

115	0.35	3000	30,750	31.27	43,000	52,000	0.12	115	100	0.67	10,000	46,000	118.0	73,000	0.07	23.3
									125	0.63	10,000	37,000	102.4	61,000	0.09	11.13
									200	0.72	10,000	42,000	102.3	74,000	0.06	.....
165	0.72	3000	50,170	50.4	45,000	62,000	0.09	165	150	1.27	10,000	68,000	151.0	68,000	0.05	20.0
									150	1.11	10,000	101,000	144.0	69,000	0.06	26.0
									200	1.00	10,000	60,000	102.0	101,000	0.07	10.0
215	0.72	3000	105,000	107.0	50,000	68,000	0.07	215	150	1.27	10,000	100,000	202.3	81,000	0.04	23.3
									150	1.21	10,000	140,000	196.0	84,000	0.05	26.6
									200	1.05	10,000	100,000	170.0	120,000	0.05	14.29

The velocity of the steam in a pipe, or the size of the pipe, is usually determined by the maximum friction loss or pressure drop which can be allowed.

Due to the low pressure drop and high velocity possible with superheated steam a greater amount of steam can be transmitted with the same pipe, so that the radiation losses per unit weight are less. Therefore, in order to make a fair comparison between the radiation losses in carrying superheated and saturated steam, not the direct heat should be considered, but the total heat losses in percentage of the heat conveyed in the steam through the pipe. Table 4 gives the heat losses in pipe lines from 4 to 12 inches in diameter for saturated steam and for steam superheated 100, 150, and 200 degrees, at gauge pressures of 100, 150, and 200 pounds. The values are calculated using the heat-transmission coefficients as given in Tables 2 and 3, and the values for saturated steam include the liquid heat of the condensed steam.

PRESSURE DROP OF SUPERHEATED STEAM IN PIPE LINES.

The question of friction loss in pipe lines carrying gases or vapors has been studied by many authorities, and a number of formulas have been established which differ considerably. The author has had the opportunity of making observations on the friction loss in long pipes with both superheated and saturated steam, and after a considerable study of the subject, has come to the conclusion that, particularly for long pipe lines and moderate pressures, the formula—

$$P = \frac{C\ W^2\ L\left(1 + \frac{3.6}{d}\right)}{V\ d^5}$$

which varies with the diameter of the pipe—using the coefficient 0.0001321 suggested by Babcock—is approximately correct for smaller pipe lines up to 4 inches in diameter; while for larger pipes, especially with wet steam, the formula $P = CW^2L/Vd^5$, with the coefficient suggested by Martin, Hawksley or Gutermuth, varying from 0.0003135 to 0.0003557 and depending upon the wetness of the steam and the surface of the pipe, is more likely to be correct.

For superheated steam with low temperatures and low velocities so that the wall temperature is lower than that of saturated steam and the walls are therefore covered inside with a film of water, the pressure drop is approximately the same as that of dry saturated steam. With higher temperatures, higher velocities and correspondingly higher wall temperatures so that the pipe remains dry inside, the coefficient C for superheated steam varies not only with the diameter of the pipe but also with the velocity, the pressure, the absolute temperature, and the density; and the friction loss agrees closely with that observed by Fritsche,* so that his suggested formula for the coefficient C which follows can be considered reliable and used safely in calculating the pressure drop for long pipe lines carrying superheated steam and moderate pressures:

$$C = 0.0000022\ (R/144)^{0.148}\ (T/pw)^{0.148} d^{-0.260}$$

where—

T = absolute temperature, degrees F.
p = absolute pressure, pounds per square inch.
w = velocity, feet per second.
d = pipe diameter, feet.
R = 85.7 for steam.

*Mitteilungen über Forschungsarbeiten, V. D. I. (Berlin), Heft 60.

In order to facilitate the use of this formula a table was calculated giving for air (R = 53.34) the values of 1,000,000 C. corresponding to values of the ratio T/pw for pipe diameters ranging from 1 inch to 48 inches. This table is given in Marks' Mechanical Engineers' Handbook. When used for steam its values must be increased by 7 per cent on account of the different value of R used.

As far as the author knows, all formulas for determining pressure drop, particularly for steam, are based on tests where the pressures in no case exceeded 300 pounds per square inch. For higher pressures they should be used with caution. In his extensive study of this subject and search for a formula which would cover all conditions, Biel* found that the pressure drop caused by the resistance to flow of any fluid, for the same diameter of pipe, the same length, same temperature, and the same constant R, can be expressed by the equation $P_d = CW^n p^m$, in which the values of the exponents are $n = 1.852$, $m = 0.852$. Approximately the same values were found by Brabbée,† who made extensive tests, particularly to determine the resistance, in pipe lines, for high- and low-pressure steam and hot-air heating. He found, however, that the exponents vary with the roughness of the pipe. For steel pipe his exponent is 0.853.

The question arises as to how far the known formulas for the resistance of flow are applicable to high-pressure and high-temperature steam. It is the opinion of the author that the pressure drop in *straight lines* carrying superheated steam of 300 to 600 pounds pressure is from 10 per cent to 25 per cent less than that calculated by the use of the existing formulas for moderate pressures.

A comparison of the results of tests made at pressures between 250 and 440 pounds with those obtained by using the Fritsche formula bears out this statement. The pressure drop in fittings and bends was carefully determined by mercury U-tubes, and the resistance to the flow, which is usually expressed in lengths of straight pipe, was considerably higher than formerly supposed. As compared with the formula of Conrad Meier, given in the paper of D. E. Foster,‡ the values obtained were from 20 per cent to 35 per cent higher.

The foregoing would tend to show that most of the pressure drop in pipe lines takes place in the bends, fittings, and valves, especially the latter, rather than in the straight pipes. In particular, sudden restriction of area and change in direction of flow affect the pressure of the steam. This has been verified by careful tests even with moderate pressures.

BEST VELOCITY OF STEAM FLOW FOR USE IN HIGH-PRESSURE LINES.

It is therefore apparent that in pipe lines carrying high-pressure steam, provided they consist of long, straight pipe sections, high velocities are permissible, and for high pressures—say, from 300 to 500 pounds—10,000 to 12,000 feet per minute seems to be the most advisable for lines over 5 inches in diameter and with a continuous steam flow, as, for instance, when the steam is used in turbines. With reciprocating engines, naturally, lower velocities are to be chosen, depending upon the length of the line, the percentage of cut-off, and the number of revolutions of the engine. For high-pressure reciprocating engines located at a considerable distance from the steam generator it was found advisable to make that portion of the steam line near the engine of a larger diameter in order to minimize the fluctuation of the steam flow due to the intermittent steam demands of the engine, and to allow higher velocities by means of a correspondingly smaller pipe diameter, near the boiler.

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* Mitteilungen über Forschungsarbeiten, Heft 44.
† Zeitschrift des Vereines deutscher Ingenieure, 1916.
‡ Effect of Fittings on Flow of Fluids through Pipe Lines, Trans. A.S.M.E., vol. 42, p. 647.

TABLE 5.—VALUES OF 1,000,000 C.

Diameter in inches	Values of the ratio T/Pw						
	0.008	0.01	0.015	0.02	0.03	0.04	0.05
4	1.35	1.38	1.43	1.46	1.53	1.60	1.65
5	1.23	1.29	1.33	1.37	1.44	1.50	1.55
6	1.16	1.21	1.26	1.30	1.37	1.43	1.48
8	1.10	1.14	1.18	1.22	1.27	1.33	1.37
10	1.03	1.07	1.11	1.14	1.19	1.25	1.29
12	1.00	1.03	1.07	1.10	1.14	1.19	1.23

TABLE 6.—PRESSURE DROP AND RADIATION LOSSES IN PIPE LINES CARRYING HIGH-PRESSURE SUPERHEATED STEAM.

Pressure Abs. lb.	Deg. Superheat	Pressure Drop	Velocity ft. per min.	Wt. Steam lb. per hr.	Million B.t.u. trans per hr.	Rad. loss	Pressure abs. lb.	Deg. Superheat	Pressure drop	Velocity ft. per min.	Wt. Steam lb. per hr.	Million B.t.u. trans per hr.	Rad. loss
4-in. Pipe (Extra Strong)													
315	100	2.60	6000	16450	17.92	32100	415	100	3.03	6000	21190	23.25	34750
	150	2.46	6000	15400	17.18	37150		150	2.84	6000	19740	22.20	39550
	200	2.34	6000	14470	16.50	42200		200	2.68	6000	18460	21.20	45250
6-in. Pipe (Extra Strong)													
315	100	1.58	6000	37300	40.60	47300	415	100	1.79	6000	47900	52.60	51350
	150	1.48	6000	34900	38.90	54750		150	1.68	6000	44600	50.20	58400
	200	1.39	6000	32700	37.40	62000		200	1.59	6000	41750	48.00	66900
8-in. Pipe (Extra Strong)													
315	100	1.44	7000	76000	82.80	60800	415	100	1.68	7000	97900	107.3	65750
	150	1.36	7000	71200	79.50	70200		150	1.57	7000	91250	102.6	75000
	200	1.29	7000	30900	76.40	79700		200	1.49	7000	85400	98.2	86000
10-in. Pipe (Extra Strong)													
315	100	1.37	8000	142100	154.8	76800	415	100	1.57	8000	183000	205.0	83250
	150	1.20	8000	133000	148.2	88700		150	1.48	8000	170400	192.0	94900
	200	1.22	8000	125000	142.7	100900		200	1.4	8000	159500	183.6	108700
12-in. Pipe (Extra Strong)													
315	100	1.65	10000	258500	282.0	91300	415	100	2.01	10000	332500	365.0	99000
	150	1.56	10000	242000	270.0	105200		150	1.90	10000	309500	348.0	112700
	200	1.47	10000	227000	259.0	119800		200	1.79	10000	290500	334.0	129200

If a line carrying high-pressure steam contains a number of bends, fittings, and valves, lower velocities than those mentioned in the preceding paragraph should be employed if excessive pressure drop is to be avoided.

The higher cost of the larger sizes necessitated by lower velocities, and the strength of the material required for higher pressures, are factors to be taken in consideration in determining the size of the line.

The table previously mentioned giving values of 1,000,000 C is intended only for low pressures. Table 5 has therefore been devised to give values of the same quantity for pressures up to 600 pounds and any superheat. Table 6 gives the pressure drop and radiation losses for pipes from 8 to 12 inches in diameter and for 300 to 400-pound pressure.

PIPE LINES CARRYING SUPERHEATED STEAM FOR OTHER PURPOSES THAN POWER GENERATION.

The question of friction losses or pressure drop in a pipe line, particularly with superheated steam, is of importance only when the steam is used for generation of power. If it is used for heating, drying, or process work, only its temperature need be considered, the pressure being of no consequence. Higher velocities can therefore be chosen. Very often the pressure has to be reduced before the steam be used, and a pressure drop in the pipe line is even desirable. With saturated steam, however, it is necessary to limit the velocity. The usual velocities with saturated steam in commercial pipe lines are about 3,000 to 6,000 feet per minute. Any considerably higher velocity increases the danger of water hammer, particularly in long lines with a number of bends, fittings, and changes in diameter, and when the line cannot be completely and continuously drained. In such lines high velocities with saturated steam cause considerable vibration, especially when *much* moisture is carried with the steam, due to the effect at higher velocities of the greater weight of water at any change in direction of the flow.

With superheated steam, particularly at sufficiently high temperatures, practically no moisture is present in the line, and nothing prevents the use of the highest speeds desired. Velocities of 12,000 to 15,000 (and more) feet per minute can be and have been applied in larger pipe lines without any undesirable effects. In figuring the total radiation losses in Table 4, velocities of 3,000 to 5,000 feet per minute were assumed for saturated steam, and 6,000 to 10,000 feet for superheated steam, which are conservative values that enable a fair comparison to be made.

Table 4 shows that the greatest saving, as far as radiation losses are concerned, is obtained with 150 degrees superheat, so that when the steam is superheated only for the purpose of elimination of condensation or reduction of radiation, a moderate superheat of 100 to 150 degrees, depending upon the length of the line, is sufficient. It does not mean, however, that moderate superheat is in general more desirable. If the steam is used for power purposes, the advantages of high superheat in the prime mover will more than overbalance the slightly higher radiation losses.

A remarkable feature is the very slight increase of pressure drop, which throughout the whole range of the table, with one exception, does not exceed 1 pound for 100 feet of pipe length, in spite of the fact that the velocity was doubled. The lower density of superheated steam, the absence of moisture, the dry pipe walls, and the reduced amount of foreign matter—the greatest part of which is usually left in the superheater—account for the low pressure drop.

It is commonly believed that even with superheated steam the pipe walls are covered on the inside with a film of moisture. This is true for low superheat with low steam velocities. At higher superheat and high velocities, however, the wall attains a temperature higher than that of saturated steam, and remains dry.

EXAMPLES SHOWING ADVANTAGES OF USING HIGHER VELOCITIES WITH SUPERHEATED STEAM.

The three following examples from actual practice demonstrate how the possibility of applying higher velocities with superheated steam may be advantageously utilized in reducing the radiation losses and the cost of the piping and covering.

The management of a large textile concern, realizing that it was necessary to lower their power cost, decided as one means to this end to avail themselves of the advantages of superheated steam. Each of the five boilers in the plant was accordingly equipped with superheaters for 200 degrees superheat.

The steam was for use in two engines, one triple-expansion and one compound. The compound engine was located about 950 feet from the boiler house. The piping between boiler and engine was entirely changed, the old 8-inch piping being replaced by a new 6-inch line. With the old piping and saturated steam the average pressure drop in the line between boiler and engine was 6 pounds. With the new line and superheated steam it was 7.5 pounds. The comparatively low pressure drop in the new line was mainly due to the appreciably decreased steam consumption of the engine. The radiation loss of the 8-inch line would naturally be 33 per cent higher than that of the 6-inch line for the same steam temperature.

A chemical company manufacturing aniline dyes had an 8-inch pipe line, 1,400 feet long, in their plant, conveying steam from their boiler room to the process house. An increase in production required the approximate doubling of the steam consumption in the power house, and a second pipe line accordingly was planned. At the same time the use of higher temperatures was found advisable, since the boiler pressure could not be increased appreciably. Superheaters were installed in the boilers, and approximately double the amount of steam superheated was sent through the pipe. Pressure drop was of no consequence, while the radiation loss per pound of steam conveyed was decreased 36 per cent.

A heating plant had two mains 10 inches and 8 inches in diameter and 2,300 feet long. The 10-inch line was used in the cold winter months, while the smaller one was utilized during the remainder of the year. With saturated steam the larger steam demand in the winter months could not be supplied with the small pipe alone, due to the danger of water hammer and excessive vibration, and the larger pipe had to be used. In order to have dry steam at the end of the line, and avoid the losses in liquid heat of the steam condensed in the line, superheaters were installed, and it was found that the greater amount of steam could be conveyed through the smaller pipe without any difficulty. The radiation loss as compared with the 10-inch line was 20 per cent less.

DISCUSSION.[1]

Geo. A. Orrok[2] said that the author had once more emphasized that if the outside surface was kept low, and arranged so that it radiated but little, the pipe-line losses would be small. He had covered the velocity of steam in pipes up to 10,000 feet per minute. This was a very ordinary velocity for power plants; in most cases the velocities were greater. It should always be borne in mind that losses were proportional to the surface and not to the amount of steam in the pipe.

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[1] These extracts from the discussion deal with those portions of the paper appearing in the preceding abridgment.

[2] Cons. Engr., 17 Battery Place, New York, N. Y. Mem. A.S.M.E.

J. A. Barnes[3] wrote calling attention to a power plant at Conneaut, Ohio, operating with saturated steam through a pipe line between 400 and 600 feet long. At this distance it was impossible to operate the engine at anything like its full capacity. Superheaters were installed and about 100 degrees of superheat was obtained at the engine, and the capacity was materially increased. At the Bellman-Brook Co., a finishing company in Fairview, N. J., superheaters were installed which gave from 100 to 150 degrees of superheat, and the output of the plant was increased approximately 35 per cent due entirely to the elimination of radiation losses.

W. H. Armacost[4] said that he had plotted some comparative curves, taking the data from papers presented before the Society by Messrs. McMillan, Bagley, Heilman, and the author. The curve plotted from the data of Mr. Broido's Table 3 bore out his statement that there was less radiation by using superheated steam in pipe lines than with saturated steam, due to the fact that saturated steam, in transferring heat to the pipe wall, partly condensed or gave up some of its latent heat, which heat was not as closely combined with the steam as the sensible heat of a liquid or the heat of superheated steam.

J. H. Lawrence[5] stated that in designing the Hell Gate station, probably the largest in the country, the losses in the steam line had been compared with the losses estimated by various formulas, and while there should have been a very excessive drop in the lines, the loss was only about half that given by any formula in the handbooks, and so small that it was almost negligible.

He did not believe in low velocities. The author mentioned 10,000 feet, but he considered this too low for most steam pipes in a large station. His concern used as a maximum 15,000, and in certain cases, where the lines would only be used in case of an emergency, they did not hesitate to go to 20,000 feet per minute. In most handbooks there were certain rules about velocity which provided for 6,000 or 8,000 feet for saturated steam and 10,000 for superheated steam. Such information was very misleading.

Their experience has shown that the velocity should increase as the size of the pipe increased. For one particular case, if a 2-inch pipe was used, the velocity would be around 2,000 feet per minute, otherwise the losses would be very excessive. With a 6-inch pipe the proper velocity would be about 6,000 feet, and so on. On the 14-inch size they figured about 14,000 feet velocity as the minimum. They had stopped there, however, as they did not wish to go as high as 20,000 feet in ordinary practice.

E. G. Bailey[6] said that the steam meter had now become such an integral part of the plant equipment for measuring the output of boilers, steam consumption of turbines and various units as well as the steam distribution, that the conditions suitable for the installation of such meters should be considered not only in the selection of the proper size of pipe but also in the layout.

One meter manufacturer used limited velocities of $V_{max} = 8{,}000\sqrt{\text{sp. vol.}}$ and another $V_{max} = 10{,}500\sqrt{\text{sp. vol.}}$ With steam at 250-pound pressure and 200 degrees F. superheat having a specific volume of 2.35 the maximum velocities would be 12,240 and 16,000 feet per minute, respectively. These velocities were capable of being metered accurately only when the steam was flowing through straight pipe of reasonable length on either side of the primary device.

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[3] Superheater Co., New York, N. Y.
[4] Engr., Superheater Co., New York, N. Y. Asso-Mem. A.S.M.E.
[5] Engrg. Mgr., Thos. E. Murray, Inc., Mem. A.S.M.E.
[6] Bailey Meter Co., Cleveland, Ohio. Mem. A.S.M.E.

L. B. McMillan[7] stated that it was a matter of common knowledge that heat was transmitted much more readily from saturated steam to a surface than from superheated steam. However, granting a large difference in transfer coefficients from saturated and superheated steam to pipe wall, the question was, how much difference would this make in actual heat losses?

The value of α for saturated steam used by the author was over twelve times that for superheated steam and it would be only natural to expect that the loss from the saturated-steam pipe would be far larger than that from a superheated-steam pipe at the same temperature. Such would be the case if the transmission of heat from steam to pipe wall were the only factor involved, but the heat must be transmitted from the outside of the pipe to the air, or through insulation to the air. Therefore the additional resistance to heat flow was so great outside of the pipe that the difference in inside resistances became very small in comparison.

In the case of bare pipes there were larger differences, but even with bare pipes the differences were of the order of only 10 per cent. These differences, however, were of little importance as few bare pipes were used for the transmission of superheated steam.

Most published tables on heat losses from bare pipes were based on loss per square foot per degree temperature difference between *pipe surface* and air. These were therefore correct for the stated condition, regardless of what was inside the pipe.

Referring to Table 4, in which a lower radiation loss per pound of steam was shown for superheated steam than for saturated steam, this result was based entirely upon the assumption that the velocity in superheated-steam lines was exactly double the velocity in saturated-steam lines. This table showed higher actual radiation losses on the superheated-steam lines, but on the basis of the doubled velocity so much more steam was passed through the superheated-steam lines that the loss per pound was shown to be less. An explanation of just why the author assumed a doubled velocity in the superheated steam line instead of some other ratio would be necessary in order to give significance to this table.

In fact, it was Mr. McMillan's understanding that the practice of calculating steam-line sizes on an assumed velocity was antiquated and that the preferred method was one based on allowable pressure drop.

L. L. Barrett[8] stated that those engineers who had occasion to consider the heat losses from bare pipe lines conveying steam would be grateful to the author for his presentation of extracts from Eberle's paper which had never before been translated into English.

Referring to the tables in the paper, the heat losses on which Tables 1 and 2 were based had been obtained by weighing the water condensed in the pipe line under test. This method was inaccurate, as stated by the author, and the later determinations of McMillan, Bagley, and Heilman were preferable. The constancy of the value of K for the covered 3-inch line in Table 2 was a good illustration of the inaccuracy of this table. Both McMillan and Bagley had shown that this value would vary considerably with the temperature difference.

The comparison in Table 4 was based on the assumption that all the condensation from the saturated steam lines was wasted. It could not be admitted that this was the general practice in power plant or marine work. In such cases it was almost the invariable practice to return the condensation to the boilers. Any other practice would be most wasteful and would result in a poor showing for the heat cycle of the plant.

It would be interesting to know on what basis the radiation losses for superheated steam in Tables 4 and 6 had been figured where the steam

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[7] Cons. Engr., Johns-Manville Co., New York. Assoc-Mem. A.S.M.E.

[8] Mgr. Engrg. Dept., Keasbey & Mattison Co., New York, N. Y. Assoc-Mem. A.S.M.E.

velocities were shown as 7,000, 8,000, and 10,000 feet per minute, as the highest steam velocity on which data were given in the paper was but 5,910 feet per minute.

Referring to the author's summary (appended to the complete paper) no data were given to support the heat-transmission coefficient between steam and pipe wall of 400 for saturated steam there mentioned. The values of this coefficient found in the experiments of Clement and Garland at the University of Illinois ranged from 1,649 to 2,740 and the value found by McAdams and Frost at the Mass. Inst. of Tech. in 1922 was 2,400.

The author's conclusion that the coefficient of heat transmission from steam to air remained practically constant up to a temperature of 350 degrees F. was disproved by the work of McMillan, Bagley, and Heilman, all of whom showed a very large increase in the coefficient between 100 degrees and 350 degrees F. in the case of bare pipe. The increase in the coefficient from 100 degrees to 350 degrees temperature difference was 60 per cent, according to Heilman's paper.

The author, in his closure, said that it was very true, as had been stated by Mr. Barrett, that the analysis of radiation losses in pipe lines in the paper was not based on the exeperiments of Messrs. McMillan, Bagley, and Heilman, all of which were made with electrically heated pipes, and the object of which was mainly to determine the efficiency of various types of pipe covering. His paper was not intended for this purpose. Its object, particularly the first part dealing with radiation losses, was to show, first, that there was a difference between saturated and superheated steam as far as imparting of heat to a pipe wall was concerned; and, second, that in comparing the radiation losses of saturated and superheated steam, not the direct heat in B.T.U. was to be considered, but the total heat losses in percentage of the heat conveyed in the steam through the pipe. As the Munich tests were believed to be the only extensive ones conducted with saturated and superheated steam flowing in the test pipe, only these tests could be considered and analyzed in the paper.

Mr. Barrett had looked at the paper only from the standpoint of pipe covering, and therefore had noticed only the data showing the radiation losses. If he would look upon the paper as an engineer and would consider steam velocities, temperature differences, pressure drops, etc., he would better realize the object of the paper and would understand the analysis presented.

Mr. Barrett objected to value of 400 given as the heat-transmission coefficient between the steam and pipe wall because other investigators had found it to be considerably higher, or up to 2,700. From Table 1 of the paper it would be seen that at a temperature difference between the steam and the outside air of 250 degrees, the heat loss per square foot of surface per hour in B.T.U. was 750. With a heat-transmission coefficient of 400, the temperature of the wall must be $750 \div 400 = 1.875$, or less than 2 degrees below the steam temperature. At a wall temperature equal to the steam temperature, the coefficient would be infinitely large. As it was very difficult to measure accurately the temperature of the steam and the pipe wall, and a difference of 2 degrees was likely to occur, the value of coefficient could not be definitely fixed. Four hundred was mentioned in this paper only as a comparison with the value of superheated steam, which was only 32 at a velocity of about 80 feet per second.

As Mr. Armacost had said, the difference between the data of the tests analyzed in the paper and those of Messrs. McMillan, Heilman, and others, was very slight.

Mr. McMillan wondered why the difference between the heat transmission of saturated and superheated steam had been gone into in such detail. As mentioned early in the paper, this subject was of considerable interest as far as the use of superheated steam for heating and drying purposes was

concerned. So far as the author was aware, the influence of the velocity of superheated steam on the heat transmission, particularly in connection with the question of whether the pipe was wet or dry inside, had never before been discussed, and was believed to be of considerable interest to engineers.

The velocities taken in Table 4 were in accordance with general engineering practice. The pressure drop was only slightly higher than that for saturated steam and, with the exception of one case, did not exceed 1 pound throughout the range of the table.

With reference to the question asked by Mr. C. W. Gordon* as to why the friction in pipe lines at high pressures was less, while in fittings it was higher than that derived by calculating according to standard formulas for moderate steam pressures, he would say that this was probably due to the fact that in fittings the steam changed its direction and a part of its kinetic energy was lost, and that this was more pronounced at higher pressures than at the lower ones.—"Mechanical Engineering," May, 1923.

THE USE OF STEAM AT EXTREMELY HIGH PRESSURES.

The value of steam as a means of transforming heat into work lies in the comparatively large amount of heat energy which can be transferred from the furnace to the engine by every pound of the working fluid without involving pressures or temperatures greater than ordinary constructional materials will stand. Practical considerations for many years limited steam pressures to a maximum of about 200 pounds per square inch, but with the improvements in materials and the development of manufacturing methods, a decided tendency has been evident during recent years to operate boilers at higher and higher pressures. A steam pressure of 350 pounds per square inch is not regarded to-day as anything exceptionally high for a large power plant and about 500 pounds per square inch is already in use in this country. There are two main reasons for this tendency to increase pressures. One is the desire to secure a greater thermodynamic efficiency of the plant, and the other is the appreciation of the practical benefits of the reduction in the size of steam pipes which the higher pressures render possible. The latter reason is of considerable importance, for when a single turbo-generator may require as much as 150 tons of steam per hour, the less the volume of the steam, so long as boiler drums, pipes and joints will stand, the better. As to efficiency, the theoretical increase with increase of pressure is not very great, and the overall thermal efficiency attained by such a station as Radcliffe, for example, with steam at only 160 pounds pressure, might well be envied by many stations working at nearly double the pressure.

The same reasons which have caused engineers to increase steam pressures to those which are being adopted in modern power-plants are impelling them to consider the employment of higher pressures still. It is officially reported that the Edison Illuminating Company of Boston, U.S.A., are arranging for the installation of a boiler and turbine to work at 1,200 pounds per square inch. The turbine will be rated at 2,000 kw., and will exhaust at a pressure of 375 pounds, the steam, after re-heating being passed on to the main turbine. A good deal has been heard recently of a Swedish design of boiler to generate steam at 1,500 pounds per square inch. The inventor keeps the steam-raising elements in continuous rotation, the idea being that the water shall be held against their internal surfaces by centrifugal force. The centrifugal pressure of the water would tend to keep the heated metal

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* Exper. Engr., Superheater Co., New York, N. Y. Jun-Mem. A.S.M.E.

surfaces thoroughly wetted and free from bubbles, while the inner surface of the water-layer would allow of the free escape of the steam made at all points along the element. Apart altogether from the obvious mechanical difficulties which have to be overcome, it is questionable whether the avoidance of turbulence of the water is altogether a good thing for heat transmission, and furthermore one would imagine that the need for the elimination of steam bubbles in contact with the walls would be diminished rather than increased with very high pressures on account of the correspondingly reduced size of the bubbles. Whether the complications inherent in a boiler of this type are justified, or whether the inventor has introduced them as a safeguard against less serious or even imaginary troubles we cannot say, but whatever the outcome of the design the departure from ordinary practice will do good by the establishment of facts.

The generator of steam at excessive pressures is bound to give rise to problems in boiler construction and maintenance which may not be easy to solve, and the question obviously arises as to what compensating advantages in efficiency may be hoped for. The value of a pound of steam is, of course, by no means proportional to its pressure, but only to the amount of its heat which can be transformed into work by expansion to some lower pressure. Since the publication of the steam tables and equations of Professor Callendar, engineers have had reliable means for computations on the properties and characteristics of steam at all pressures and temperatures. Taking 0.5 pound absolute, or a vacuum of about 29 inches, as the lowest practical exhaust pressure, a pound of dry steam at 500 pounds pressure, if used in an ideally perfect engine will only do 15 per cent more work than the same weight of steam at 200 pounds pressure, used under the same conditions. The proportion of the heat supplied to the steam which can theoretically be turned into work is 35.3 per cent in the former case and 31.96 per cent in the latter, so that the gain either in work or efficiency due to the higher pressure is not very great. If we push the initial pressure still higher, say to 2,400 pounds per square inch, much the same result is obtained. The available work per pound, calculated from Callendar's tables, is 261.16 pound-centigrade units, or only 27.05 per cent more than in a pound of steam at 200 pounds pressure, and the thermodynamic efficiency of a perfect engine using steam even at this extremely high pressure would not exceed 41.78 per cent.

So far we have been considering steam in the dry saturated condition, but in practice, of course, greater efficiency is sought by superheating it to the highest temperature which practical considerations permit. This may be taken, at present, as about 700 degrees F. In order to avoid interpolation and to take advantage of the greater convenience of the Centigrade scale, we will assume an upper limit of temperature of 380 degrees C. (= 716 degrees F.). With steam at this temperature and a 200 pounds pressure utilized as assumed above, in a perfect engine exhausting at 0.5 pound absolute, the available heat will be 251.53 units, which is 33.82 per cent of the total heat supplied to the engine. At an initial pressure of 500 pounds the available heat will be 276.58 units or 37.6 per cent of the total, and at a pressure of 2,400 pounds the corresponding figures are 295.78 units, or 43.28 per cent of the total. These figures are sufficient to show how slowly the value of steam increases with its pressure, whether superheated or not. Nevertheless every increase in efficiency, however small, is worth obtaining, if it is not counterbalanced by corresponding disadvantages, and the tendency in modern power station practice is, as we have already stated, in the direction of continually higher pressures.

One of the boldest experiments yet attempted in the use of extra high pressure steam is now being carried out at Rugby, and appropriately enough in the Willans Works. There the English Electric Company, Limited, are putting down a plant, to the order of the Benson Engineering Company,

Limited, of London, in which steam will be generated at the extraordinary pressure of 3,200 pounds per square inch, which will be used to drive a turbine designed and constructed in the works in question. Before entering the turbine the steam will be throttled to 1,500 pounds per square inch by means of a reducing valve and its temperature raised to 700 degrees F. by superheating. Before considering in detail the whole cycle of operations involved, we may refer to the properties of steam at the pressure mentioned, because in them lies the reason for the experiment. As the pressure in a boiler is raised the steam of course becomes denser and denser, and equally of course the water becomes less dense owing to its expansion by heat. A continuation of the process must result in the steam and water arriving at a state of equal pressure, temperature and density, or in other words being indistinguishable from one another. This occurs when the pressure reaches 3,158 pounds per square inch, the corresponding temperature being 374 degrees C. (705 degrees F.). The separation surface between the water and steam vanishes at this point and the substance can be called either water or steam with equal accuracy.

The importance of this phenomenon lies in the fact that in any boiler working at a pressure over 3,158 pounds per square inch water can be changed into steam without any surface of separation. The steam is not evaporated from the water, but the latter, with which the vessel may be solidly filled, simply becomes steam throughout its mass when the temperature becomes high enough. The latent heat has vanished and the temperature of the high pressure feed water rises continuously until it has the expansive properties of steam. The Benson boiler has therefore no need for any steam-drum, which is a matter of great importance in view of the pressure worked at, but partakes more of the nature of an economizer in which the water is steadily heated in its progress from the inlet to the outlet. It would not be profitable to discuss the design in detail from such particulars as are available, and the plant is indeed too far from completion to render any description of its component parts otherwise than premature. The fundamental principle embodied is the production of steam at a pressure above the "critical point" and how this may be utilized when obtained is of secondary importance.

The cycle of operations proposed at Rugby and the theoretical efficiency of the process, as calculated by Professor Callendar, are as follows: Water at 100 degrees F. is heated to a temperature of 706 degrees F. and a pressure of 3,200 pounds per square inch. The volume will then be 0.052 cubic feet per pound and the heat required for the operation 820 B.T.U. per pound. The steam is then throttled under conditions of constant total heat, to a pressure of 1,500 pounds absolute, the temperature falling to 596 degrees F. It is next heated at 1,500 pounds pressure to a temperature of 700 degrees F., the specific volume being then 0.380 cubic feet per pound, and the total heat rising from 890 to 1,310 B.T.U. The next step is to expand it adiabatically in a high pressure turbine to a pressure of 250 pounds. This will result in steam at a temperature of 401 degrees F., with a wetness of about 8 per cent, the heat available for driving the turbine being 166 B.T.U. per pound. The steam is then reheated at 250 pounds to 700 degrees F., increasing the specific volume to 2.68 cubic feet per pound, and the total heat from 1,144 to 1,375 B.T.U. per pound. An adiabatic expansion in a second turbine down to a vacuum of 28 inches renders a further 425 B.T.U. available for work. Work equivalent to nearly 10 B.T.U. is required to pump the water back into the boiler at 3,200 pounds, thus completing the cycle of operations.

The heat available for work in the two turbines is 571 B.T.U. and the total heat supplied is 1,471 B.T.U., so the overall efficiency is theoretically 38.8 per cent. The low pressure turbine alone, working with an available heat of 425 B.T.U. out of a total of 1,305 B.T.U., is working on a cycle

with a theoretical efficiency of 32.5 per cent, and doing this, moreover, without the use of pressures or temperatures greater than are permissible in current practice. The margin in theoretical efficiency by adopting the whole of the cycle discussed, does not appear very striking, but it is nevertheless sufficient in Professor Callendar's opinion to promise a profitable advance in practice, if the boiler, superheater and turbine do not give serious trouble. More than half the heat, as he has pointed out, would be added to a practically homogeneous fluid in a boiler without steam space. The Rugby plant is designed for a total output of 1,000 kw., so that the high-pressure turbine will be a very small one quite apart from the limitations imposed by the steam pressure at which it works. Supposing that it had an efficiency ratio of only 50 per cent, it would absorb only 83 B.T.U., passing on an equal amount to reduce the work of the intermediate superheater. With this efficiency ratio for the high-pressure turbine, the steam leaving it would be dry instead of wet, thus minimizing the danger of blade erosion. As the low-pressure turbine is only working under conditions within the range of experience, its performance can be accurately forecasted, and it is indeed not proposed to construct this turbine at all at the outset.

Whatever the result of the Rugby experiment, Mr. Benson is to be congratulated on his courage in attempting so marked a change from accepted practice as regards the generation of steam. That there are possibilities in his method is undeniable, but it is equally certain that the process will present its own peculiar difficulties. Not the least of these, we imagine, will be that of throttling a hot fluid with a density of 0.052 cubic feet per pound from 3,200 pounds to 1,500 pounds pressure. He is well into the region of hydraulic pressures, with temperature difficulties super-added. The governing of a turbine working with a pressure of 1,500 pounds per square inch is also a matter which requires consideration. The governing problem, moreover, will become much more difficult when the low-pressure turbine is added. Intermediate steam heating involves the presence of a comparatively large amount of steam in the heater and its connections. This steam has passed the governor valve and is, so to speak, out of control, and ready to drive the low-pressure turbine without reference to anything the main governor is doing. Close governing is therefore anything but easy. This difficulty has, we believe, been experienced acutely in certain plants with intermediate re-heating and is exercising the minds of designers. It is not therefore peculiar to the Benson plant, and no doubt will be overcome in this case as in others.—"Engineering," June 22, 1923.

BOOK REVIEW.

MARINE WORKS, BY ERNEST TATHAM. D. Van Nostrand Co., 8 Warren St., New York City. Price $5.00, net.

A book primarily for Civil Engineers, but of interest to all those connected in any way with maritime matters. It deals with all forms of docks, basins, etc., for handling ships and, in addition, describes salvage operations in great detail. Also the problems involved in keeping rivers navigable are covered thoroughly.

ASSOCIATION NOTES.

The Society has been requested to participate in The American Marine Congress to be held at New York City in November, 1923, and will be represented in the General Committee of the Congress. The following are the Society's representatives.

Rear Admiral J. K. Robison, U. S. Navy.

Rear Admiral R. S. Griffin, U. S. Navy, Retired.

Commander C. A. Bonvillian, U. S. Navy.

Lieutenant Commander F. W. Sterling, U. S. Navy, Retired.

The Congress has been called by the American Marine Association and it is hoped that substantial progress will be made toward a definite and satisfactory movement to benefit shipping in this country.

The following members have joined the Society since the publication of the last JOURNAL:

NAVAL.

Abercrombie, David, Ensign, U. S. Navy.
Baker, H. Martyn, Ensign, U. S. Navy.
Birk, Joseph W., Lieutenant, U. S. Navy.
Bolles, Harry A., Ensign, U. S. Navy.
Busby, Percival W., Lieutenant U. S. Navy.
Chase, Harry T., Ensign, U. S. Navy.
Cleveland, Lloyd K., Lieutenant, U. S. Navy.
Cunningham, Winfield S., Lieutenant, U. S. Navy.
Fowler, Wendall C., Ensign, U. S. Navy.
Gelm, George E., Captain, U. S. Navy.
Hickey, Stockard R., Lieutenant, U. S. Navy.
Maeser, Earl, Ensign, U. S. Navy.
Moody, Ralph G., Lieutenant, U. S. Navy.

Moran, Henry G., Ensign, U. S. Navy.
Russell, William J., Lieutenant, U. S. Navy.
Wells, Frank O., Lieutenant, U. S. Navy.
Wood, McFarland W., Ensign, U. S. Navy.

ASSOCIATES.

Burt, Arthur G., care Consolidated Gas Electric Light Power Co. of Baltimore, Baltimore, Md.

Keeler, Earl A., care Brown Instrument Co., Wayne & Windrim Ave., Philadelphia, Pa.

ADDRESSES.

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